Vol. 12, No. 5 September-October, 1955

LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

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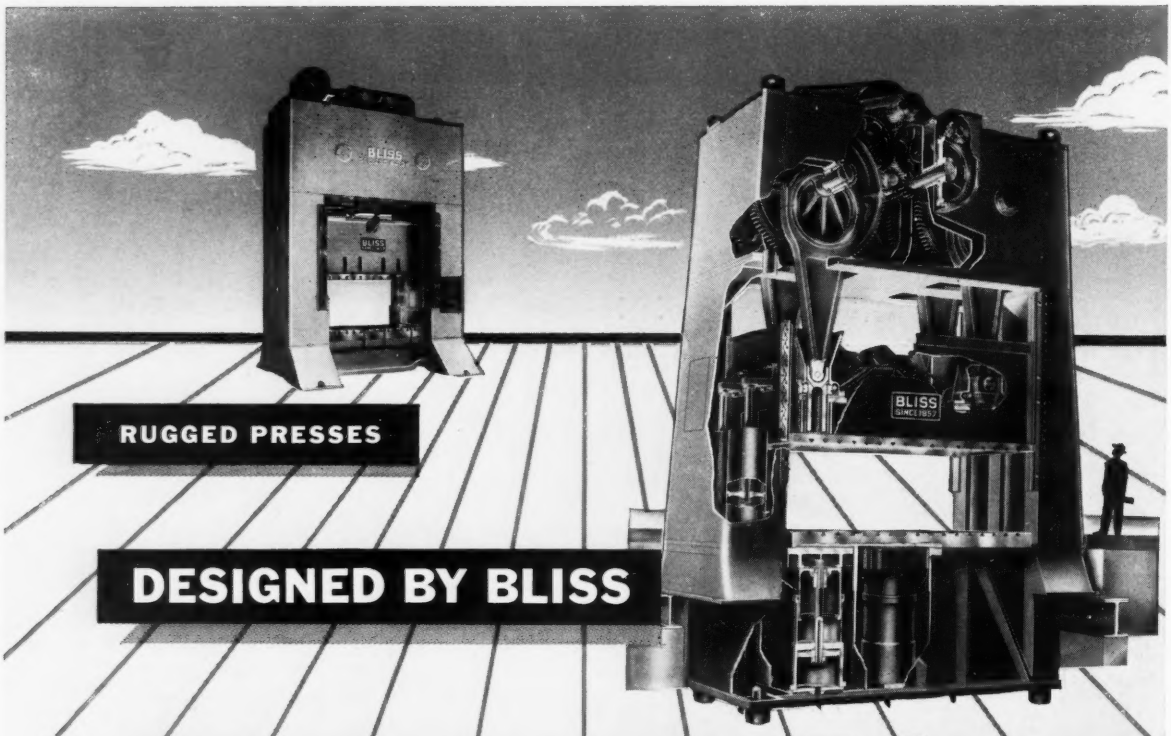
JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS



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LUBRICATION ENGINEERING

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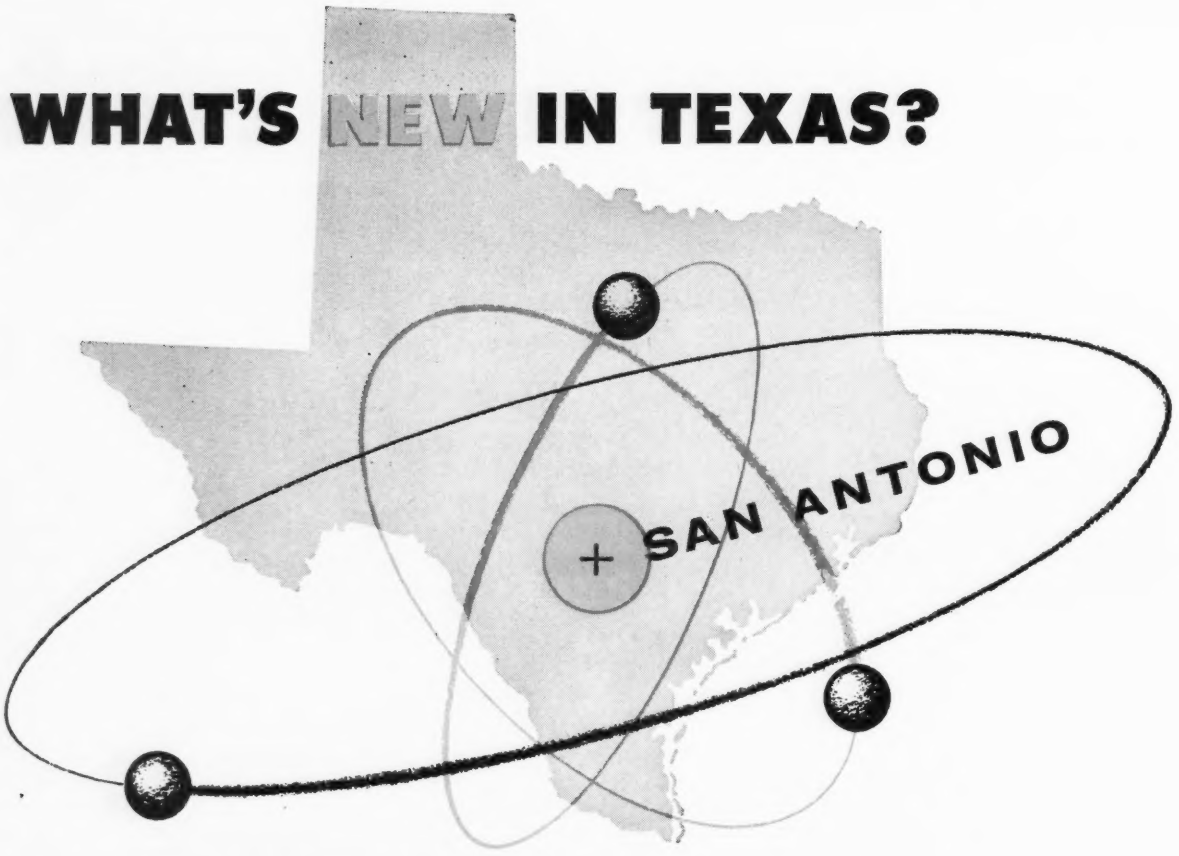

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Meet the Authors

Meet the authors of the papers appearing in this issue of **Lubrication Engineering** (Vol. 11, No. 5). Mr. F. R. Archibald is the author of the series of biographies entitled "Men of Lubrication"; his article in this issue covers the life and accomplishments of A. G. M. Michell. Messrs. W. C. Milz & L. B. Sargent, Jr., and M. B. Peterson & R. L. Johnson, are co-authors of their respective papers.

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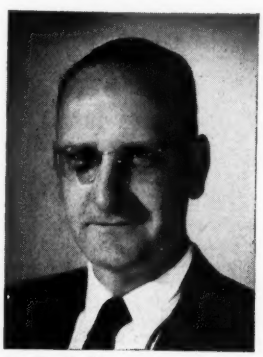

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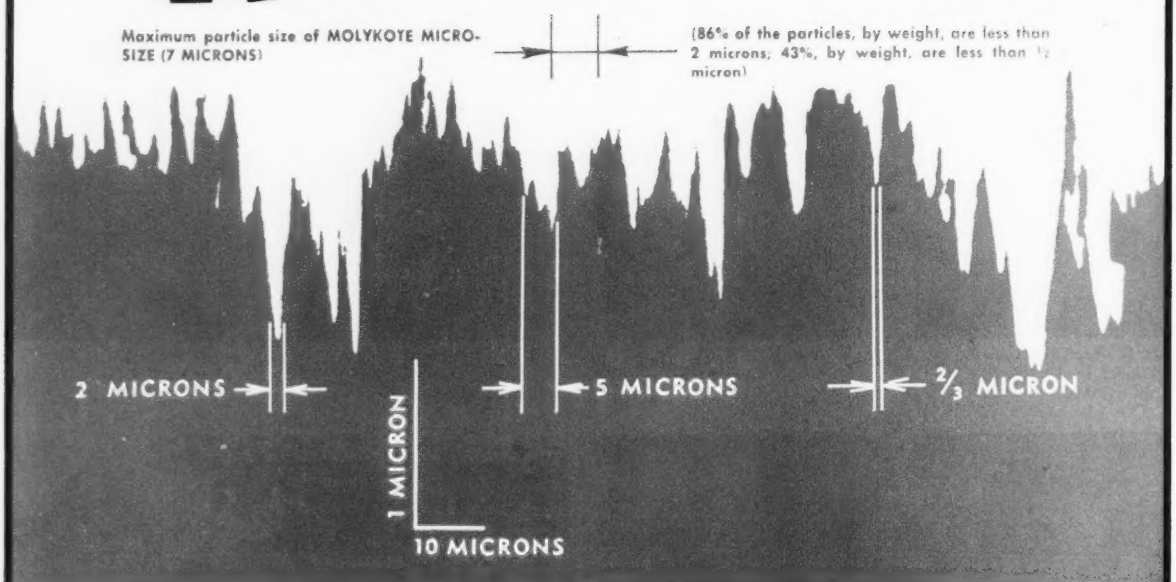
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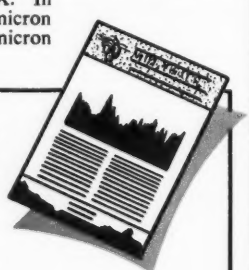
The lower portion of the illustration shows in magnified cross-section a ground steel surface (finish 8 micro-inches R.M.S.). Taper sectioning was employed for study purposes resulting in a vertical magnification 25 times the horizontal (reference: "Mechanical Wear" edited by John T. Burwell, Jr.). Horizontal magnification, 1000X; vertical magnification, 25000X. In the illustration 1 mm equals 1 micron horizontally; 25 mm equals 1 micron vertically.



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Note: Information on MOLYKOTE appears in Sweet's Product Design File 1955, Section 3C.

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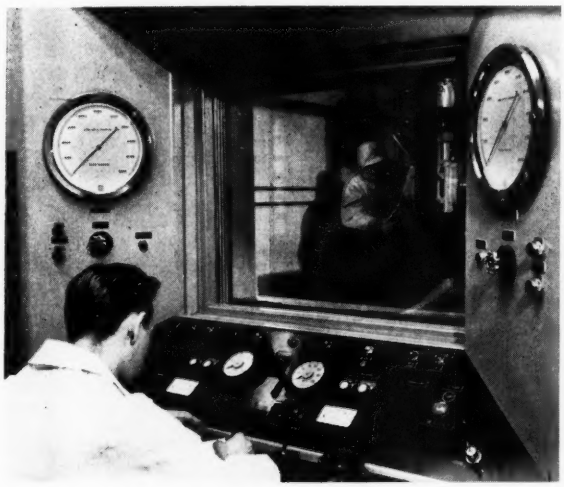

New Journal Bearing Research Laboratory. The American Brake Shoe Co. has recently completed a new $250,000 journal bearing laboratory at their Research Center in Mahwah, New Jersey. (A general view of the laboratory is pictured above.) It comprises three rooms on two floors as follows: a test room containing a fully instrumented bearing, a control room, and a machinery room.

The test room can be held at any desired temperature from minus 40 F. to 125 F., plus and minus 2 degrees. Thermo-pane windows on two walls of the room allow observation during tests, and a large insulated access door is provided for installation or removal of the heavy machinery. Approximately in the center of the test room is the test machine which consists basically of a power unit, a driven axle having standard railroad journals and associated equipment on each end, and provision for placing loads on the journal bearings. The power unit consists of two DC motors coupled with a magnetic clutch and rigged to drive the test axle by means of a chain. The smaller is a 5 horsepower gear motor rated at 15 to 75 rpm and is used for starting the test axle and for running it up to speeds equivalent to train speeds of six miles per hour. Above this speed the second motor of 60 horsepower can be connected through the magnetic clutch to drive the axle at speeds up to 115 miles per hour. The test axle is mounted in two large main support bearings which rest in concave seats and are self-aligning. The journals are replaceable. Loads on either journal box can be varied up to 20,000 pounds per journal, the maximum for railroad journals. The actual weights for loading the test journals are located in the machinery room. Loads and torques can be measured readily. The loads can be raised to 40,000 pounds per journal bearing by hydraulic loading.

The main components in the machinery room are two refrigeration compressors, condenser, two dead weight loading pendulums, two motor generator sets to supply DC power for the motors in the test room, hydraulic oil system, and a lubricating system for the main support bearings on the test axle. A 40 horsepower motor is used for driving the larger refrigeration compressor.

All the equipment in the test room and machinery room is controlled at the operator's panel in the control room. Test results are recorded continuously for starting torque, running torque, load, speed, time, direction, test bearing temperature, and journal box oil temperature for both journal bearings under test. A microphone is hooked into a public address system so that the operator can communicate with workers in either the test room or the machinery room.

Economics of Lubrication. The stocking, dispensing, and handling of lubricants lends itself to economic study. At Kodak Park about $55,000 worth of lubricants a year are purchased, and it costs about $275,000 to apply, wipe up, and dispose of them. It is estimated that before a lubrication program was put into effect, based on the present size of the plant, the purchase cost would have been $150,000 a year, and the application cost well over $1,000,000. Many smaller plants can make proportionate savings by utilizing to the fullest extent the talents of those engaged in the lubrication business. (C. L. Pope, Eastman Kodak Co.)

Volatile Corrosion Inhibitors. Volatile corrosion inhibitors represent the latest approach to rust prevention. These are organic chemicals that sublimate slowly and deposit an invisible film of rust inhibiting agent on surfaces to be protected. Greatest industrial use has been in the packaging of ferrous metal spare parts where these inhibitors have simplified packaging, storage, and shipment. This form of protection permits the immediate use of packaged items without proceduces normally associated with the removal of liquid or solid surface coatings before placing the equipment into service. The method is particularly advantageous where complicated mechanisms are involved.

An attractive property of volatile corrosion inhibitors is their ability to reach all exposed surfaces by diffusion through the atmosphere. Only small quantities of the inhibitor are necessary to protect large surfaces; it has been calculated that condensed films only five molecules thick are sufficient to provide rust inhibition. However, an inhibitor easily deposited by condensation can vaporize just as readily, leaving surfaces unprotected. For this reason the inhibitor vapor must be maintained above a minimum concentration, a requirement not difficult to meet in small confined spaces such as engine cylinders, packaged equipment, or even moderate sized drums and tanks. Systems involving large volumes may require special precautions. Excessive moisture, or free acids or alkalis also create problems.

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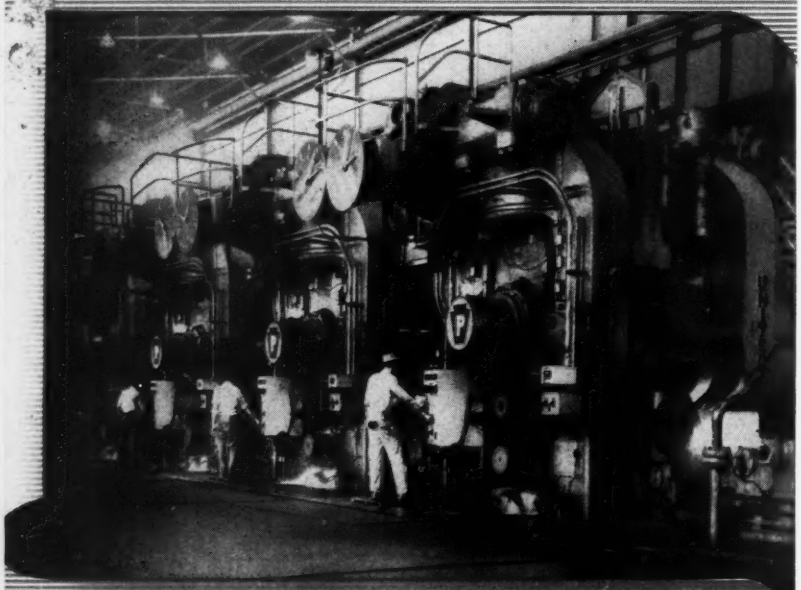

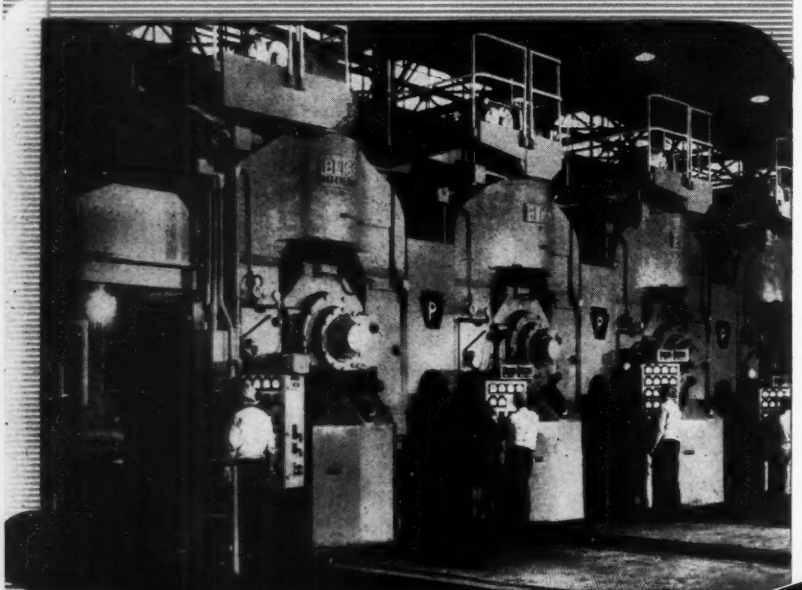

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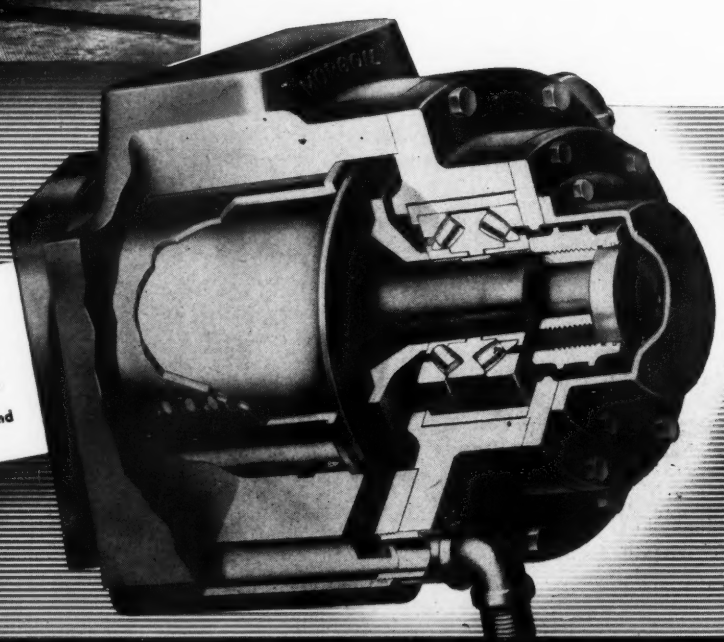

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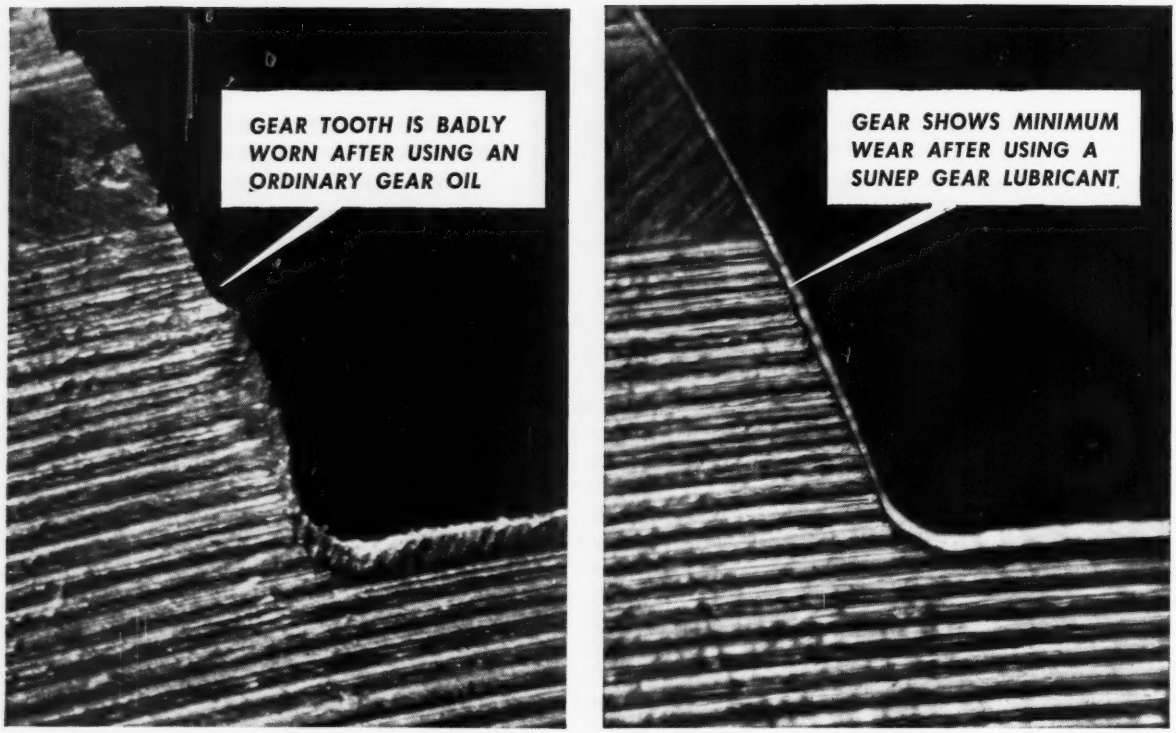

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An Empirical Correlation Between Dynamic & Kinematic Viscosity, by Dr. H. E. Mahncke (Westinghouse Research Labs., East Pittsburgh, Pennsylvania).

When an engineer or designer is called upon to calculate bearing performance or other quantities involving viscosities, it is necessary for him to convert the units in which viscosity is usually specified (either kinematic or arbitrary units) into absolute or dynamic units. This is usually done by a relation such as the following:

$$\mu = \nu d = (0.22t - \frac{1.80}{t})d \qquad (1)$$

where μ = absolute viscosity (centipoises), ν = kinematic viscosity (centistokes), t = Saybolt viscosity (seconds), and d = density (gms. per cc.).

There are practical disadvantages to the use of such a relation. It is tedious and sometimes confusing to apply. The density data are not generally available and empirical rules which are relied on for density calculation do not always give the required accuracy.

The density data is not difficult to determine in a modern laboratory and we have therefore adopted the practice of measuring it concurrently with our viscosity determinations. The procedure is as follows:

Kinematic viscosities are determined with the Ostwald Cannon type capillary viscometer as described under ASTM Designation D445-46T. Under good conditions this gives a reproducibility and an accuracy of ± 2 parts per thousand. Densities are measured by weighing a charge of oil into a standard Cassia flask, which is a 110 ml. volumetric flask with 10 ml. graduated to 0.1 ml. on the neck. This is suspended in the same constant temperature bath as the viscometer and a density temperature curve may be obtained at the same time as the viscosity curve. Densities measured in this way are accurate to one part in a thousand.

In the course of accumulating data of the above sort, we have observed an interesting empirical correlation between absolute and kinematic viscosity in which, conveniently, density does not appear. It takes the form:

$$\mu = A\nu^B \qquad (2)$$

and thus is convenient for conversion of kinematic to absolute viscosities when density data are not available. In the present case this was established by measuring the viscosities of fifty samples of oils, ranging in viscosity from 5 to 1300 centistokes at 40 C. (105 F.) over the temperature range of —12 to 150 C. (10 - 300 F.). This entailed a total viscosity variation of 1 to 100,000 centistokes and a density variation of .740 to .945 gms. per cc. The data (some 500 points) was found to fit a straight line when plotted on a logarithmic scale. Least squares analysis gave the following values for the constants:

$$\mu = 0.8\nu^{1.017} \qquad (3)$$

with an average deviation of 2.03%, a maximum deviation of 8.4%, and a standard deviation of 0.13%. The seemingly high maximum deviation is caused by our rounding off the observed viscosities to one decimal place, which introduces a larger error in the lower viscosities.

A spot check of the Harvard data[1] on pressure-viscosity, shows that the above relationship applies to pressure as well as temperature variation. In this data a density range of 0.74 to 1.04 and a temperature range of 0 to 218 C. (32 - 425 F.) is encountered.

It should be emphasized that this is an empirical correlation and is only offered as such. Materials which are not chemically similar to mineral oils, such as silicones and fluorocarbons, will not obey this relationship although another one of the same type can be developed.

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(1) "Pressure Viscosity Report." ASME Research Publication (1953), The American Society of Mechanical Engineers, 29 W. 39th St., N. Y. 18, N. Y.

ACKNOWLEDGEMENT. The author wishes to express his sincere appreciation to Dr. Ruth Goodman, of the Laboratories, for carrying out the statistical calculations.

Solid Lubricants For Titanium, by M. B. Peterson & R. L. Johnson, (NACA Lewis Flight Propulsion Laboratory, 21000 Brookpark Rd., Cleveland 11, Ohio).

The lubrication of titanium has consistently proven to be a difficult problem. NACA experience with other metals has indicated that solid lubricants will effectively lubricate those metals on which they will form adherent films. MoS_2 or graphite applied with organic bonding agents have been most generally recommended for lubricating titanium.[1] The halides, CdI_2, $CdCl_2$, and PbI_2 (which have shown promise as solid lubricants[2]), are believed to have greater chemical affinity for titanium than either MoS_2 or graphite. As a matter of interest, the more effective fluid lubricants for titanium include halogenated hydrocarbons.[3]

The relative ability of the solids to lubricate titanium was investigated in a low speed friction and wear apparatus. The apparatus and the experimental procedure are described in Ref. 2. The apparatus is basically 3 hemispherically tipped (3/16-in. radius) rider specimens loaded (40 lbs total) and sliding (5.7 ft per min) in a continuous track against a horizontal flat disk. Both rider and disk specimens were made of commercially pure titanium RC-70 (99 + percent Ti < 0.2 percent C). All lubricants except graphite were used in dry air. The disks were ground, vapor blasted, and carefully cleaned. A copious supply (¼-in. thick) of the powdered solid was placed on the disk. Runs were made without benefit of a bonding agent so that the inherent ability of the

(Continued on p. 299)

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(Continued from p. 297)

solid lubricants to form adherent films would be accentuated. The duration of each run was 30 minutes. Friction, wear, and surface conditions after operation were determined.

The effects of running time on friction coefficient for powdered MoS_2, CdI_2, $CdCl_2$, PbI_2, moist graphite, and grade 1010 petroleum oil containing 5 percent benzyl chloride are shown in Fig. 1. Since dry graphite would not lubricate, it was therefore run with moisture present. Variation in moisture probably caused the more erratic friction (± 0.03) observed with graphite. At the end of the 30 minute runs, continuous films of lubricant were apparent on the surface of the specimens lubricated with CdI_2, $CdCl_2$, PbI_2, and moist graphite. More wear had taken place with moist graphite than with the other successful lubricants. Lubrication failure accompanied by severe galling occurred when either MoS_2 or the oil were used as the lubricant.

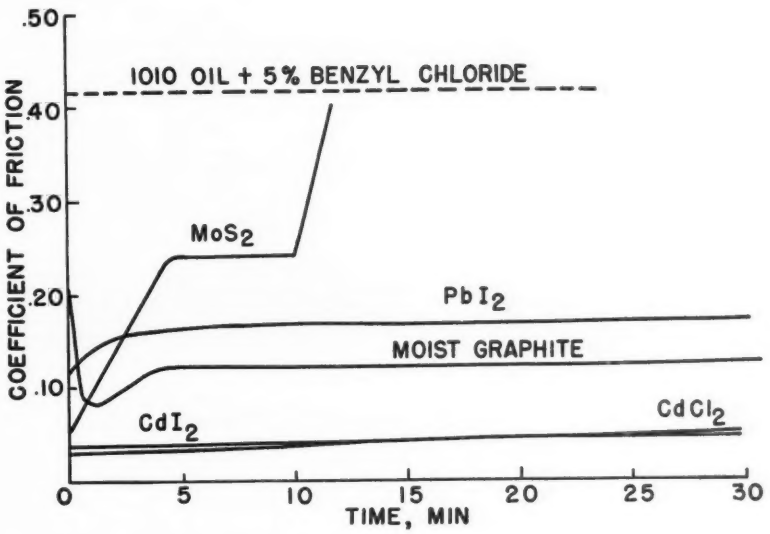

Fig. 1. Friction of titanium surfaces with various lubricants. Speed, 11 rpm; load, 40 lb; atmosphere, dry air.

These preliminary data indicate that $CdCl_2$ or CdI_2 may be useful lubricants for titanium alloys. Further investigation is essential to determine their effectiveness under more practical conditions.

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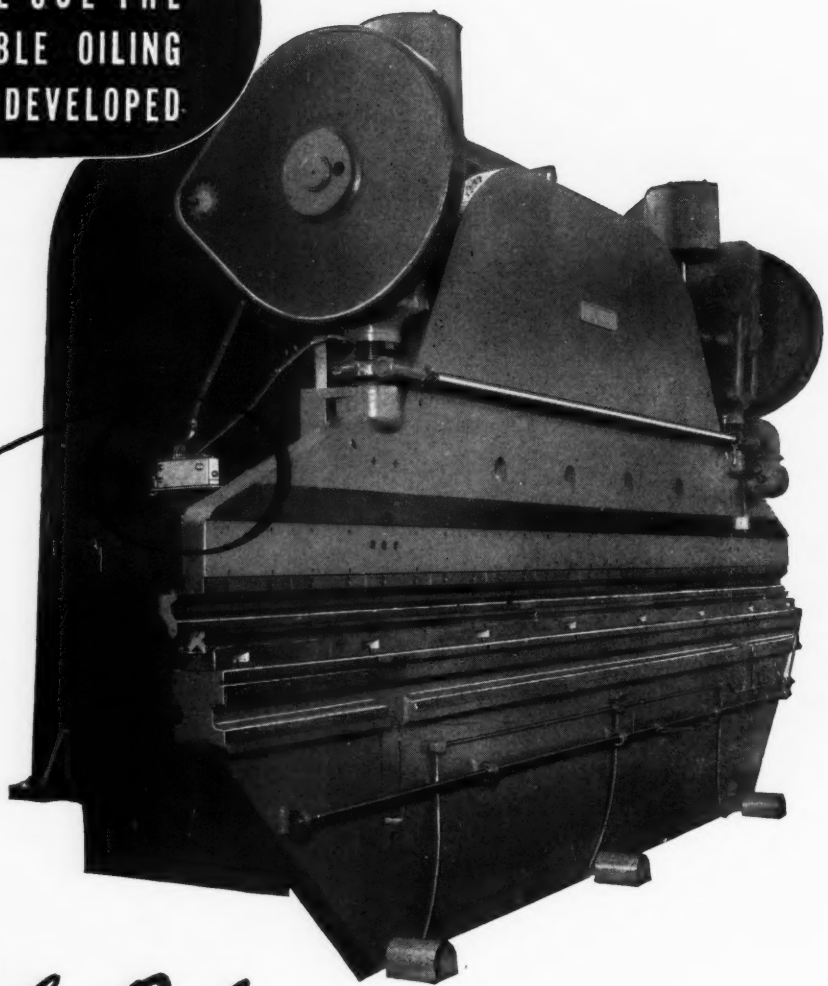

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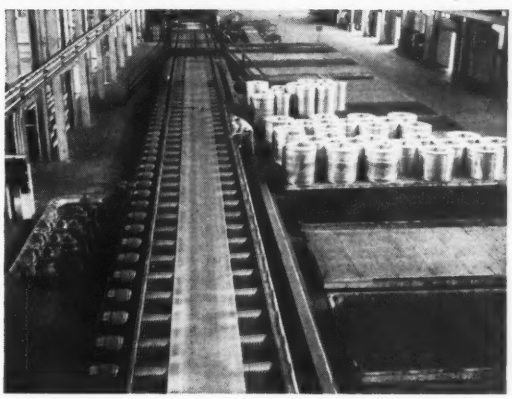

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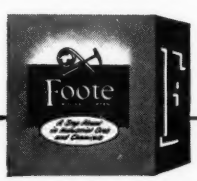

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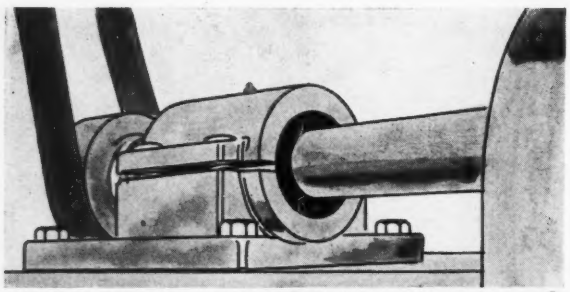

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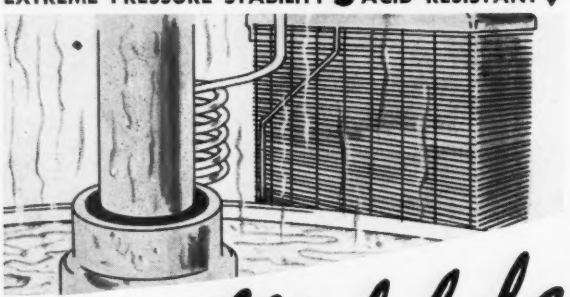

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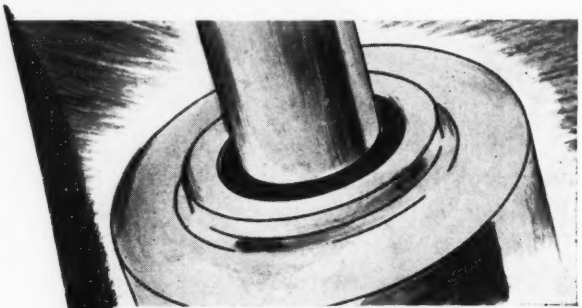

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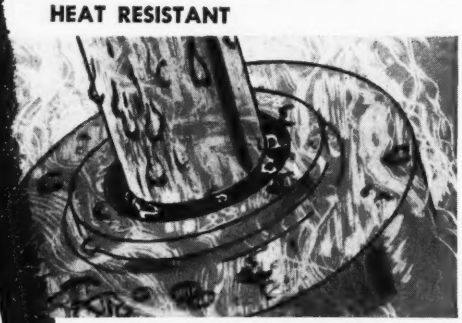

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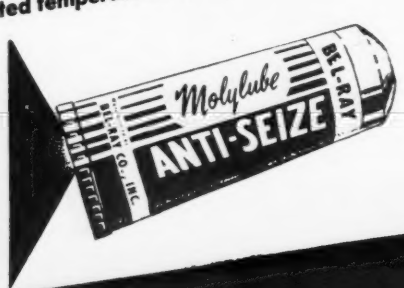

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A. G. M. MICHELL

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Men of Lubrication

ANTHONY G. M. MICHELL. Anthony George Maldon Michell was born in London, England, in 1870. His parents were originally from Devonshire but had emigrated to Australia during the fifties to seek their fortune in the newly discovered gold fields. A. G. M. Michell was born during a visit of his parents to England. The name Michell suggests a French origin but the family had apparently been in the West Country of England for many generations.

Michell grew up and was educated in Australia and in England. He spent one year at Cambridge University, but his engineering degree was obtained in 1895 at the University of Melbourne. Concurrently with his university work he obtained engineering experience in the office of the consulting engineer, Bernhard A. Smith, and in the workshops of Johns and Waygood, hydraulic and structural engineers. In 1897 he received the degree of Master of Science. His first paper, on a mode of elastic instability of beams, was published in the Philosophical Magazine at this time.

Bernhard A. Smith was concerned with hydraulic engineering and the design and construction of hydraulic turbines. Michell later became a partner in the firm. During this period the Smith-Michell regenerative centrifugal pump was invented. In this pump the fluid, emerging from the impeller, is directed on a turbine wheel connected through gears to the pump shaft, thereby reclaiming a considerable fraction of the kinetic energy of the issuing stream. Many such pumps were built for irrigation service in Australia.

After serving a year as examiner in the Victoria patent office, Michell opened his own consulting engineering office in 1903. His interest was in pumps, hydraulic turbines, and hydroelectric power plants.

The increasing size of hydraulic turbines in hydroelectric developments at this time was forcing the attention of engineers on the serious shaft thrust problem associated with their operation. Michell's activities in turbine design brought him in immediate contact with this difficulty which, undoubtedly, was fast becoming a major obstacle to the enlargement of units. Accordingly, he turned to Reynolds' theory of lubrication to find some way around the seeming impasse. Reynolds had worked out the case of the inclined flat slider, neglecting the effect of side leakage. Michell now set about the practical problem of finding what load would be supported by a slider of finite width if account is taken of the effect of side leakage. His solution to this problem is given in his famous paper, "The Lubrication of Plane Surfaces." The paper appeared in Zeitschrift für Mathematick und Physik, pp. 123-137, in 1905 and is written in English. The mathematical problem solved by Michell, in taking account of side leakage, is difficult and the computations are so lengthy that only a few reductions of some special cases have been made. In recent years the problem has been re-examined by M. Muskat, F. Morgan, and N. W. Meres, with some improvement in the results, but it remains a formidable mathematical exercise. The solution has been valuable in many ways. For a long time it was the only lubrication problem that had been solved taking account of the effect of lubricant flow perpendicular to the direction of motion. The effect of this flow is to reduce materially the load capacity calculated on the assumption of no such flow. A square slider for given operating conditions develops only 44 per cent of the load calculated on the basis of no side leakage. This is a drastic correction, and a theory which fails to allow for it can hardly be considered adequate. By using Michell's solution, correction factors have been devised for the journal bearing problem which has not been solved when side leakage is taken into account, though of recent years numerical methods have given valuable results.

Michell invented the tilting pad bearing and obtained a British patent for it in 1905. The scheme is essentially that which had been arrived at by Albert Kingsbury at about the same period. There was at one time apparently some uncertainty about the matter of priority of the invention, but both inventors have since made the history of their work public and the priorities have been well established. Michell's own statement of the case, which is worth giving, is authoritative and undoubtedly fair: "In this connection, the author avails himself of the opportunity to explain that the pivoted thrust bearings known in America and Europe respectively as the "Kingsbury" and "Michell" bearings are in principle the same; and to state that the development of these bearings was effected in its early stages by Professor Kingsbury and himself independently in their respective countries without knowledge of each other's work. Professor Kingsbury's work was commenced a few years earlier, though his first publication on the subject was later than the author's." ("Progress of Fluid-Film Lubrication," by A. G. M. Michell. Trans. ASME, Sept.-Dec. 1929, pp. 153-163.)

Although the bearing was invented in the first instance by both men for use in hydraulic machinery, its use on shipboard to take propeller thrust has been an equally important application. Michell later applied the pivoted pad principle to the journal bearing, and this construction has definite advantages in certain cases. He also applied the pivoted pad thrust bearing idea to the design of a crankless engine in which the bearing pads transmit the piston thrust to a swashplate. A number of these engines have been built and operate successfully. The arrangement has not, however, become very popular. He also invented a number of other devices of value. One of these of interest in the field of lubrication is his Workshop Viscometer. This instrument is composed of a one inch diameter steel

(Continued on p. 346)

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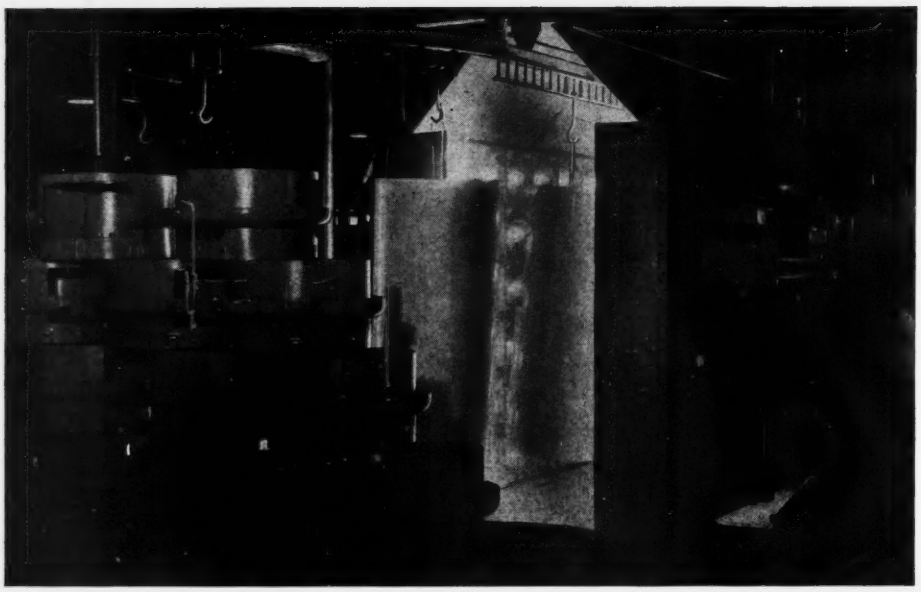

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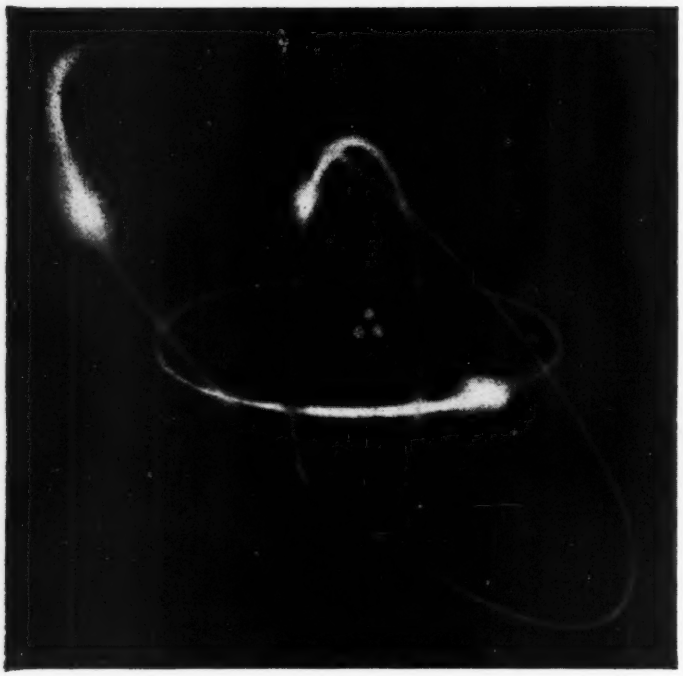

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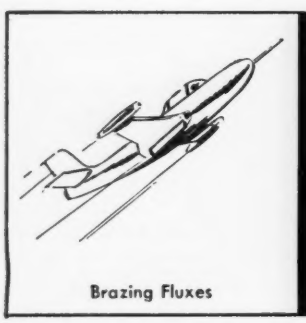

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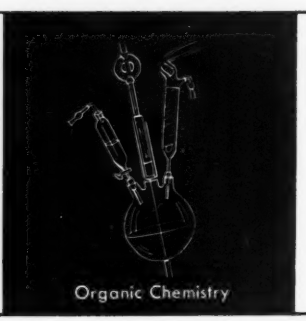

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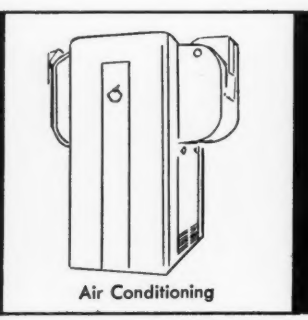

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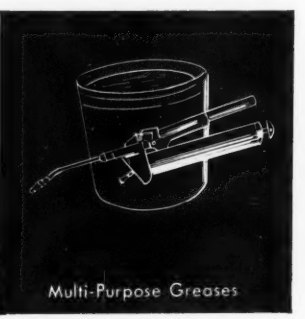

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Soluble Oil—Handle With Care

by A. A. Manteuffel*

Soluble oils are perhaps the most delicately balanced of the compounded industrial oils. Their wide acceptance in the metal cutting industry is well known and their beneficial effects in specific applications has been disclosed quite thoroughly. The fine quality product that reaches the consumer is the end result of careful compounding and handling. However, troubles can arise. This paper is intended to focus attention on those problems that can occur from the time of compounding to the time of application at the machine. Examples of potential trouble in the blend and in the emulsion are discussed in light of general known composition; and consideration is given to possible solutions.

Much has been written and discussed regarding the nature and the uses of water-based cutting fluids. Their versatility has been well acknowledged in the field of metal turning and grinding. Many excellent papers have been published which cover such phases as: the methods of application,[1] the precautions in handling, and even the scholarly study of the organisms that dwell within the emulsion.[2]

There is one important phase of the general subject of soluble or emulsified oil that has been bypassed and merits consideration. Compounders of soluble oil are keenly aware of the necessity for strict adherence to formulation and processing procedure to insure a quality product. As a result they have developed tests for checking and predicting product quality. However, this does not preclude the possibility that conditions encountered in transportation, storage, and handling could render unstable an otherwise satisfactory material. The exceptional soluble oil problems encountered by field personnel will be discussed and an attempt will be made to explain the reasons for failure of the blend or of the subsequent emulsions.

WHAT ARE SOLUBLE OILS? In dealing with a problem such as this there are certain assumptions or definitions which must be agreed upon. A soluble oil will be considered a complex blend of mineral oil and additives which will emulsify in water. Let us consider that the total blend is made of two main component portions (mineral oil and emulsifiers) which are linked together with a third ingredient referred to as a coupler. The mineral oil portion is usually a low viscosity neutral of solvent or conventionally refined origin. This portion may also contain additives other than emulsifiers selected to impart special properties such as extreme pressure or oiliness. Occasionally applications will require a higher viscosity mineral oil. The emulsifier portion usually contains petroleum mahogany sulfonates which may be supplemented with resinates, sulfonated fats, or various soaps which can be prepared by saponification with alkali metal hydroxides, ammonia, or organic amines. This portion will also contain water. The coupling agent is required to link together otherwise incompatible ingredients. Alcohols, glycols, or ethers are usually used for this purpose. The blend is very delicately balanced and will have physical and chemical characteristics such as neutralization number, viscosity, sulfated ash, etc. which are fairly restrictive for any specific formulation. This blend when emulsified in water is dispersed throughout the water which is considered the dispersing phase of the emulsion.

It is possible to formulate soluble oils for specific applications by controlling the amount of emulsifier and the types of additives. Specific soluble oils are available for normal metal turning, for heavy duty work, for grinding, for hard and soft water, and for applications too numerous to mention.

STORAGE PROBLEMS OF SOLUBLE OILS. (A) Gelling. The problems of the blend that will be considered are those that might be encountered from the time of manufacture to the time the emulsion is prepared. One of the most troublesome problems encountered with a soluble oil blend in storage is that of gelling, which becomes most apparent at low temperatures. Fig. 1 shows a typical example of gelling of the soluble oil blend. In this condition the product has a grease-like structure which can vary from semi-liquid to a rather tough consistency. This condition can be

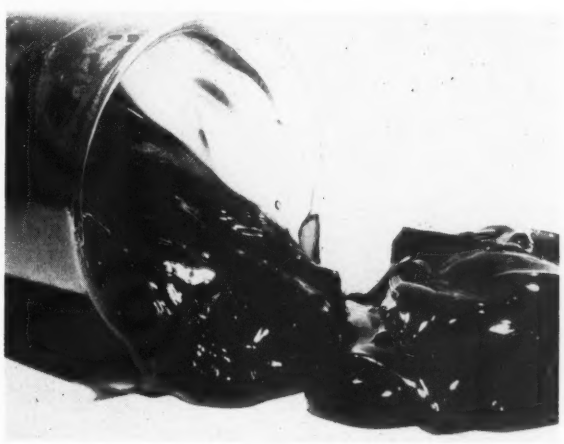

Fig. 1. Gelled soluble oil blend. This condition is often associated with low temperature storage and can result from improper moisture balance.

*The Pure Oil Co., Research & Development Laboratories, Crystal Lake, Ill.

(This paper was sponsored by the ASLE Technical Committee on Fluids for Metal Working, and presented at the ASLE 10th Annual Meeting, Chicago, April 15, 1955.)

caused by too much soap or by the use of certain types of sulfonates. However, it is usually caused by a lack of moisture in the formulation which allows a build-up of the soap crystal structure and the subsequent formation of a grease.

The loss of water can result from: (1) Storage at elevated temperatures in either sealed or unsealed containers; (2) Excessive breathing of semi-closed containers at room temperature; (3) Storage of small quantities in large containers; (4) Defective equipment which causes excessive aeration or heating during transfer from tank car or drum to plant storage.

The replacement of water in the soluble oil blend is difficult without elaborate equipment and without first determining the required amount on a small portion. The quantity of water required to rebalance the product depends, of course, on the specific formulation. Therefore, a less technical means of handling should be considered first. By warming the gelled soluble oil carefully to the extent of resolving the gel, the blend may be sampled to determine the quality of its emulsions. If the gel-resolved-to-liquid blend is non-emulsifiable, the addition of approximately 0.01% oleic acid or triethanolamine should be made in small increments until an acceptable emulsion can be formed. This will require adequate transfer and blending facilities.

(B) Phase Separation. Although stratification or phase separation of a soluble oil is difficult to observe in tank or drum storage, it is very apparent in glass storage due to a difference in color between the upper oil-rich phase and the lower emulsifier-rich phase. (See Fig. 2) When stratification occurs, the color and the viscosity of the upper portion will differ from those of the lower portion. There may or may not be an apparent difference between the quality of the emulsion prepared from the upper phase and from the lower phase. In any case the emulsions will be inferior to those prepared from the uniform well balanced oil.

This phase separation can be caused by contamination introduced from improperly cleaned tank cars or drums. It can also be caused by a deviation from the neutralization number established for the specific blend. Excessive water, insufficient emulsifier, improper choice of coupler, or inadequate mixing can also cause this type of separation. In the field a cure can usually be effected by the incremental addition of either a fatty acid or an organic amine, such as oleic acid or triethanolamine respectively.

(C) Skinning. Skinning is the formation of a "blanket-like" layer that develops on the surface of some types of soluble oil that have been exposed to the atmosphere for long periods of time. This condition is caused by the evaporation of water and by oxidation at the air-oil interface. The condition can be aggravated by warm, dry air contacting the surface of the oil. Storage in tanks and drums where breathing is allowed can also cause skinning. Obviously, it is necessary to avoid the type storage that will permit the development of this undesirable condition. If this phenomenon occurs repeatedly, it is possible to have subsequent phase separation. Thus, the observation of skinning in storage can serve well as a warning of excessive surface evaporation that must be checked. It should again be stated that this is not prevalent with all types of soluble oils.

This brief discussion of the major difficulties that can befall the soluble oil blend leads into a consideration of those problems of the emulsion which can be attributed to water or to other blend deficiencies.

PREPARATION OF THE EMULSION. The condition of the water can be as important in the preparation of an emulsion as the soluble oil itself. Factors such as extreme hardness, dispersed solids, dissolved gases, bacteria and even pH can influence the final emulsion. After the proper forethought has been given to the quality and the quantity of the water it is axiomatic that the soluble oil is added to the water and mixed thoroughly until a uniform emulsion persists.

Some formulations do not disperse as readily as others and require more thorough mixing. The ease of dispersion of the soluble oil in water is a desirable feature, particularly in applications where adequate mixing facilities are not available. Fig. 3 illustrates the difference between a good and a poor dispersing product. The ease of dispersion can be influenced by the interfacial tension characteristics of the soluble oil which can in turn be altered drastically by the addition of any one or more of the many "surfactants" available on the market. The nature of the water can also influence ease of dispersion. The judicious selection of soaps and emulsifiers by the compounder for a specific formulation is an important factor in the ease of dispersion. It should be noted that the temperature of the oil and of the water also influences the ease of dispersion at the time of emulsification. Cold soluble oil in cold water will not disperse as readily as warm oil in warm water. The level of performance for any given formulation cannot be predicted from the ease of dispersion. Entirely different factors are involved for each of these functions.

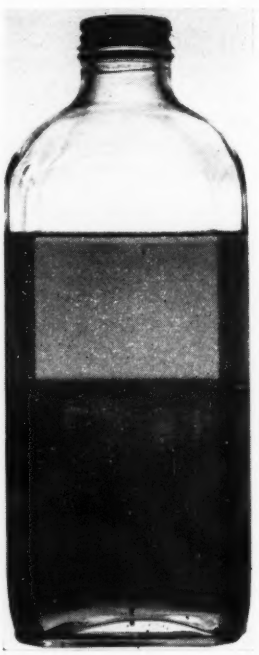

Fig. 2. Stratified soluble oil blend. (Top portion) Oil rich phase; (Lower portion) Emulsifier rich phase.

Non-emulsification might be considered an extreme of poor dispersion. This can result when an oil is add-

ed to contaminated water or to excessively hard water for which the product has not been formulated. Needless to say this condition can also result when the chemical balance within the soluble oil has been disturbed through contamination, loss of water, changes taking place in formulation during transportation as well as poor formulation. The last factor may be influenced by the type of mineral oil, the viscosity of the mineral oil and even the polarity of the mineral oil.

A perfectly uniform emulsion is usually considered the criterion of a good soluble oil since "visual" uniformity is an indication of active ingredient uniformity. Thus oil separation or creaming (the formation of a concentrated emulsion layer at the top of the total emulsion) are in most cases indicative of separation or instability in the blend. Some applications cannot tolerate this condition; others will permit it to a considerable degree. A series of emulsions which illustrates the delicate chemical balance of a soluble oil is shown in Fig. 4. The variable in this blend was the coupler. The emulsion of the preferred formulation is shown in the bottle which is fifth from the right. To the right of this emulsion is a series of four emulsions prepared from blends containing increasing incremental amounts of coupler. There is a definite increase in creaming. The series to the left of the preferred blend was prepared from blends with decreasing incremental amounts of coupler. Oil separation and creaming increase as the amount of coupler is reduced. Note the very narrow range of the perfect or the acceptable emulsions.

The above symptoms and causes of the emulsion problems have been related because the majority of them can be directly attributed to deficiencies of the blend. Some, however, can be traced to the deficiencies of the water. Correction of these shortcomings must be dealt with through adjustments in the original formulation or special treatment of the water as the case may be. Each is a study in itself and cannot be considered at this time, but should be mentioned.

Some malfunctions of the emulsion such as excessive foaming and bad odor can be corrected by adjustments in the water phase as well as in the oil phase of the emulsion. The problem of foam may very readily arise with the use of a soluble oil that has been formulated to cover a wide range of water hardness especially when it is used in very soft water. Reducing the wide differences in interfacial tension between the water and the oil usually eliminates the foaming problem. This can be accomplished by adding to the water phase of an emulsion, components which will increase the hardness of the water such as calcium acetate, calcium carbonate, etc. or by the addition of a surface active agent to the soluble oil blend prior to emulsification.

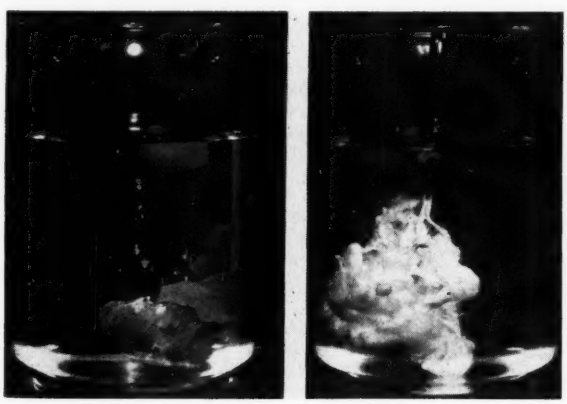

Fig. 3. Dispersion properties of soluble oil. Pouring soluble oil into water, good ease of dispersion eliminates the need for elaborate mixing equipment. (Left) Poor dispersion; (Right) Good dispersion.

The scope of the odor problem in soluble oil emulsions is quite broad and has in itself been the topic of many papers. Foul odor is associated with bacterial growth in the emulsion. This bacterial contamination must occur during use, since soluble oils as manufactured and delivered to the consumers are essentially sterile. It should also be noted that bacterial growth appears to promote stain tendencies on fine-ground iron surfaces.[3] In passing, it is sufficient to say that the odor problem stems primarily from poor housekeeping and the obvious approach is through cleanliness on the part of the consumer. Chemicals (such as chlorinated phenols) added to the oil or to the water have been effective in suppressing the bacteria which develop in the "emulsion culture". Discretion must be used in selecting the proper bactericide so that the quality of the emulsion is not disturbed regardless of whether the addition is made to the water or to the oil. The beneficial effects are usually not of long duration. In some applications merely aerating the emulsion is adequate to accomplish the same end as that imparted by the addition of bactericides. However, in other cases aeration may actually promote growth of bacteria.

The discussion related above is aimed primarily toward the personnel involved in the transportation, storage, and preparation of the soluble oil emulsion. The compounder, however, thinks of these potential problems in terms of deficiencies in one or more of the components in his product. Thus in summary these inadequacies may be reviewed in light of each separate component part of the blend.

While an excess of emulsifier in a blend will rarely cause trouble (other than increasing cost or gelling); deficient amounts will result in excessive creaming, oil separation, or non-emulsification and will narrow the allowable range of the other components in formulation. This in turn imposes limitations and requires utmost precision during compounding to obtain a satisfactory product.

Excessive water in the blend will cause separation of component parts whereas insufficient water will cause poor ease-of-dispersion as well as gelling under cold temperature storage conditions. The type and quantity of coupler are exceptionally important since excessive or insufficient coupler can cause creaming and oil separation in the emulsion as well as separation of components in the blend.

In order to insure a consistently high quality of

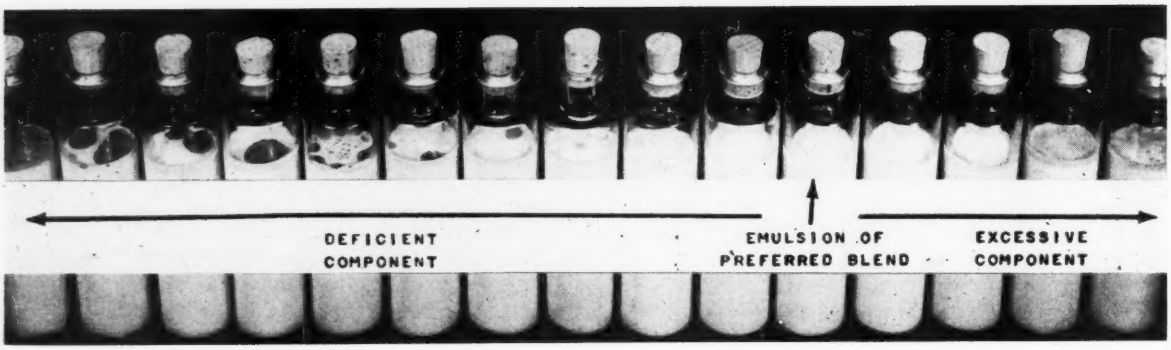

Fig. 4. Possible variations of emulsions from one blend. Incremental changes of one component in the blend can impart a wide variation in the final emulsion.

emulsion the allowable range in acidity or alkalinity for any given soluble oil formulation is rather narrow. Any deviation from this range can result in poor emulsification as well as blend separation.

Almost any type of mineral oil or petroleum product can be emulsified and stable blends can be readily prepared. In choosing the mineral oil, the gas oil, or whatever the component may be, it is important to consider the polarity, aromaticity, and acidity of that specific component.

Much is known and recorded about the colloids and emulsions. Many excellent books deal with the problems and theories involved. There is a growing awareness of the physical and chemical aspects of the subject and there has been an increasing emphasis to treat the preparation and handling of soluble oils more as a "science" rather than as an "art".

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ACKNOWLEDGEMENT. The author wishes to acknowledge the help of Mr. G. R. Cook of The Pure Oil Research & Development Laboratories for his constructive criticism and help in preparing the figures.

Lubrication Abstracts

(Editor's Note: Presented below are the first in a series of abstracts of translations of Russian articles on lubrication, by Henry Brutcher. A rare combination, Mr. Brutcher not only is an accomplished metallurgist, but translates ten different languages; since 1926 he has made available to clients in the United States over 1,200 vital scientific and production papers from inside the Soviet Union, as well as over 2,600 papers from around the world. Copies of his translations are available, at the indicated prices, from P. O. Box 157, Altadena, Calif.)

Spread in the Rolling of Flats from Steel Wire, by A. Pomp, H. Hohle & W. Leug; "Stahl und Eisen," Vol. 58, 1939, pp. 937-943; 16 figs., 3 tables, 4,500 words (Order No. 645, Price $4.50). Steels studied: Electric furnace and O. H. steels with .50-1.1% C. Rolling schedule. Spread as function of lubrication . . . and distribution of percent reductions among passes. Diagram for determining initial wire diameter for flats of given dimensions.

Selection of Lubricants for the Cold Rolling of Metals & Alloys, by I. Pavlov & Ya Gallai; "Kachestvennaya Stal," Vol. 6, 1938, pp. 34-43; 14 figs., 5 tables, 7,800 words (Order No. 899, Price $7.80). Basic requirements to be met by lubricants used in the cold rolling of metals, chiefly steels, with special reference to the various lubricants in use at the present time and their properties.

Effect of Lubrication upon Spread in Rolling, by I. Pavlov & Ya Gallai; "Metallurg," Vol. 12, 1937, Nos. 9-10, pp. 107-111; 5 figs., 2 tables, 1,300 words (Order No. 1219, Price $1.65). Contradictions in earlier experimental work on the effect of lubrication upon spread in rolling. Explantion of these contradictions on the basis of two series of experiments.

Method for Measuring the Electromotive Force in Friction & Wear, by M. L. Goldovskii; "Zavodakaya Laboratoriya," Vol. 15, 1949, No. 3, pp. 380; 1 fig., 350 words (Order No. 2336, Price $1.00). Apparatus developed by author for measuring e.m.f. created in static friction between metals. Operation apparatus; preparation of surface of friction for micro-studies of phenomena of friction and wear with "dry" and lubricated metals.

Effect of Lubricant on Physical Condition of Surface Obtained in Rolling or Drawing of Metals, by A. K. Chertavskikh; "Zhurnal Teknicheskoi Fiziki," Vol. 14, 1944, No. 9, pp. 555-560; 4 figs., 2 tables, 2,500 words (Order No. 2156, Price $4.75). Study of mechanism of decreasing lubricating value of hydrocarbons containing no active hydroxyl or carboxyl groups. Test data on performance of various lubricants on (2% C, 12% Cr) steel-copper, steel-zinc, steel aluminum, and steel-tin pairs of specimens. Lubricants studied: Gasoline, kerosene, transformer oil, petrolatum oil, crude petroleum, monohydric and trihydric alcohols. Study of wetting capacity (on aluminum) of lubricants.

Increase in Strain Hardening of Metals During Periodic Deformation in the Presence of Surface-Active Lubricants, by T. Yu Lyubimova & P. A. Rebinder; "Doklady Akademii Nauk SSSR," Vol. 63, 1948, No. 2, pp. 159-162; 4 figs., 1 table, 1,350 words (Order No. 2404, Price $2.40). Systematic

(Continued on p. 351)

Frictional Characteristics Of Plastics

by W. C. Milz & L. B. Sargent, Jr.

The frictional characteristics of seventeen plastics have been investigated under various conditions of load, speed, and lubrication. Teflon and several filled Teflons were found to have exceptionally low and nearly equivalent coefficients of friction. Some of the filled Teflons performed as well when lubricated with water as with oil. Of the other plastics, polystyrene had "dry" friction values at high speed equal to those of Teflon type plastics, and polyethylene's coefficients of friction were as low as Teflon's when using oil lubrication. Other individual performances of the plastics — as well as speed, load, and lubrication effects of all— are discussed.

In industry there are many actual and potential applications for plastics, especially in the field of bearing design, where knowledge of their frictional characteristics would be of benefit. Many investigators have reported on the exceptionally low coefficient of friction of Teflon[2, 4, 6, 7] (trade name of polytetrafluoroethylene product) and the generally good frictional characteristics of nylon[5, 6, 7] (trade name of a polyamide product). The frictional properties of other plastics such as Kel-F (trade name of polytrifluorochlorethylene product), polyethylene, methacrylate, and polystyrene have also received considerable attention. However, in most instances the reports have covered comparatively few materials tested at very low rubbing speeds with no lubrication. As a result, there has been a tendency to assign one coefficient of friction value to each plastic and neglect the possibility. of significant changes in performance caused by variations in operating conditions. Therefore, the purpose of this paper is not only to present frictional data obtained under comparable conditions on a number of plastics but also to show some of the interesting effects caused by changes in speed, load, and lubrication.

APPARATUS. Friction forces were investigated by means of a modified crossed cylinders apparatus.[3] As shown in Fig. 1, the essential elements of the apparatus are a lathe with a special frame mounted on the cross feed, and a recording potentiometer. The driven horizontal test cylinder is mounted between lathe centers, and the lathe drive and tool feed mechanisms are utilized when conducting tests. The special frame carries a nonrotating vertical test cylinder, a friction measuring transducer or strain gage, and a flexure plate-loading device for forcing the vertical cylinder against the horizontal cylinder. The vertical specimen traced a helical path on the horizontal specimen in all tests so that a new surface was continually being encountered. Dial markings on the loading gage permit application of loads within limits of ± 0.015 pounds. The recording potentiometer amplifies and records the friction force, in pounds, between the crossed cylinders as indicated by the strain gage.

The vertical specimen is part of an assembly supported by two steel flexures. This specimen is free to move downward against the restrictive action of the flexures in proportion to the friction developed between the crossed cylinders. The downward movement affects the friction transducer which is directly connected to the recording potentiometer. The system is calibrated in such a manner that the total horizontal distance across the potentiometer chart represents a friction force of two pounds. The entire loading and measuring assembly is frictionless so that the force indicated by the transducer is truly the friction force between the specimens uncomplicated by any extraneous forces.

PREPARATION OF SPECIMENS. The plastics were tested as 1" diameter horizontal test cylinders. The materials are listed in Table I and were used in commercially available rod form. The surfaces of the specimens were finished as smoothly as possible under ordinary lathe practice without using lubrication. Although there was no available instrument for measuring the surface finishes of these comparatively soft materials, their surfaces looked smooth, were smooth to the touch and free of tool marks. After machining, each specimen was carefully washed in a concentrated solution of an aryl alkyl sulfonate,[2] thoroughly rinsed with distilled water, shaken to remove excess water, and

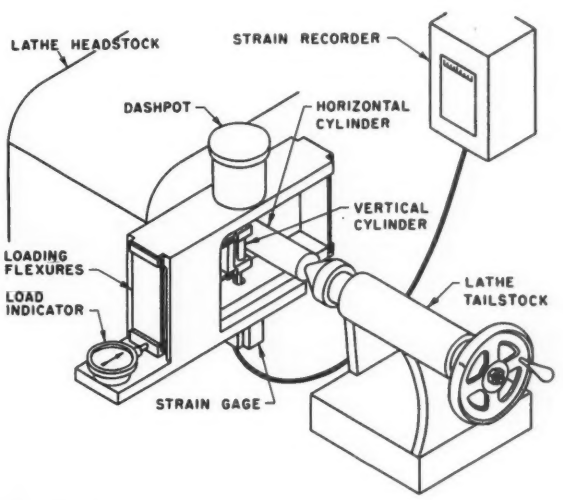

Fig. 1. Crossed cylinders friction apparatus.

*Aluminum Co. of America, Aluminum Research Labs., New Kensington, Pa.

(This paper was sponsored by the ASLE Technical Committee on Lubrication Fundamentals, and presented at the ASLE 10th Annual Meeting, Chicago, April 14, 1955.)

finally dried in a dessicator containing activated alumina. The plastics were usually resurfaced and recleaned only when changing type of lubrication. Exceptions were nylon and Nylatron G (trade name of graphite-filled nylon product) which required frequent resurfacing when run under dry conditions because of unsatisfactory reproducibility when making repeat runs over the same wear path.

The vertical specimen in each test was a ¼" diameter 1% carbon tool steel cylinder finished to a surface roughness of 1-2 microinches RMS (Profilometer). These specimens were cleaned by successive washings in benzene and acetone. During the water-lubricated tests, the steel specimens had to be repolished and recleaned several times because of their tendency to develop rust films.

LUBRICANTS. Distilled water was used in the water-lubricated runs. A well-refined uncompounded mineral oil having a viscosity of 58 SSU/100 F., viscosity index of 36, and neutralization value of 0 was used for the oil-lubricated runs. This oil was applied both with and without the addition of 5% butyl stearate, an oiliness additive.

PROCEDURE. Seventeen plastics were run against the 1% carbon tool steel cylinder, using applied loads of 1, 3, and 5 pounds; speeds of 8, 16, 52, 115, and 367 SFM; dry, water, oil, and oil plus 5% butyl stearate lubrication. Some of the plastics were physical mixtures of Teflon and powered materials, such as graphite, MoS_2, copper and asbestos. The components of the mixtures are identified in Table I.

Oil lubrication was applied by brushing onto the plastic specimen whereas water was applied in a continuous stream at the contact point with the aid of a medicine dropper. This difference in procedure was necessitated by the lack of wetting of the plastics by water when brushed on the specimens. The vertical steel specimen was indexed between tests so that no two runs were made on the same spot. Friction force measurements, which were automatically traced by the recording potentiometer, were converted to coefficients of friction by dividing the friction forces by the applied loads. A typical record of friction forces is shown in Fig. 2.

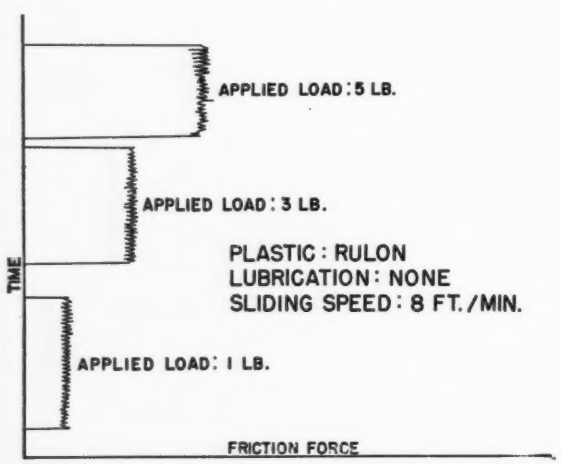

Fig. 2. Typical recordings of friction forces.

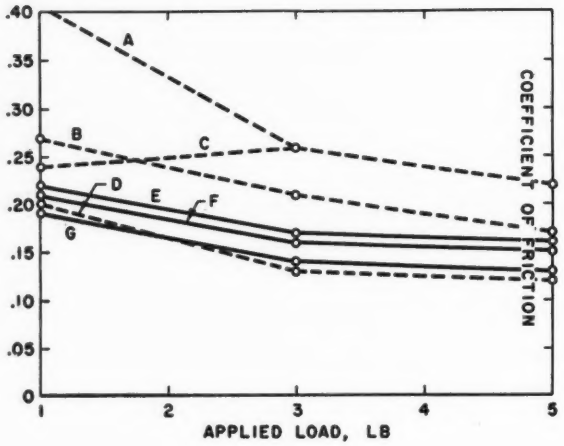

Fig. 3. Effect of load at 367 surface ft/min. using no lubrication. Key: A, Ethyl Cellulose; B, Polyethylene; C, Nylatron; D, Polystyrene; E, Teflon MoS_2 (3:1); F, Teflon Asbestos (3:1); G, Rulon.

To assist in evaluating the significance of different levels of coefficient of friction, the repeatability of results was established. This was done by performing a second, and sometimes a third run on specimens which had been resurfaced between runs. In some instances there was an elapsed time of several weeks between the initial and repeat runs. As a general rule, the coefficients of friction on dry specimens could be repeated within ± 0.01 up to a value of 0.20, and within ± 0.02 for the reported values above 0.20. For lubricated specimens, the repeatability was well within these limits.

RESULTS. For the purpose of this discussion, the first eight plastics listed in Table I, including Rulon which is the trade name of a mineral-filled Teflon, will be referred to as the "Teflon-type materials" or Teflon-type plastics. Although Kel-F is similar to Teflon, it will be classed here as a non-Teflon type material.

Some of the significant friction data are given in Table II, which indicates the maximum and minimum coefficients of friction for the plastics when tested under dry and oil lubricated conditions. The minimum values for nylon and the Teflon compositions occurred at the lowest testing speed, i.e., 8 surface feet per minute. Minimum values for all other materials were found at the highest test speed of 367 surface feet per minute. These data clearly illustrate the fallacy of identifying a plastic with any one coefficient of friction, especially when considering only the dry material. They also emphasize the importance of comparing coefficients of friction of materials only when they have been obtained under comparable testing conditions.

Considering individual performances, the Teflon-type plastics generally have the lowest friction values. The superior frictional characteristics of Teflon have been attributed to shearing at or near the surface of the material.[1, 4, 6] For most other

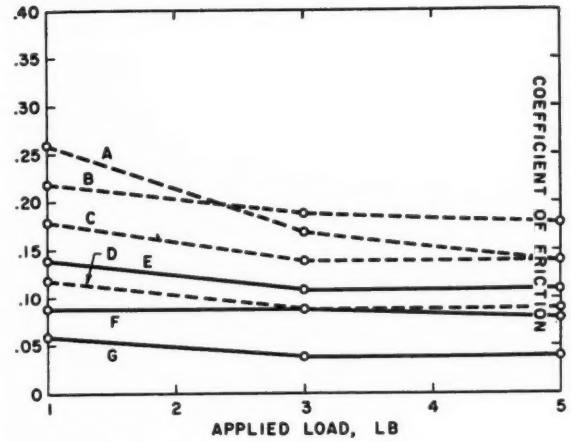

Fig. 4. Effect of load at 367 surface ft/min. using water lubrication. Key: A, Methacrylate; B, Kel-F; C, Nylon; D, Cellulose Acetate; E. Teflon Copper (1:1); F, Teflon; G, Teflon MoS_2 (1:1).

types of plastics, other investigators have shown that shearing occurs within the bulk material, thus resulting in higher friction.[1, 4, 6]

There is comparatively little difference in frictional characteristics among the Teflon-type materials. This might well be expected because Teflon itself has an extremely low coefficient of friction, and the effect upon friction of adding fillers is most likely masked by an ever-present layer of the plastic. Since Teflon has outstanding resistance to scuffing, wear, and seizure, it will resist uncovering and exposure of the embedded fillers. Therefore, good frictional characteristics are retained even though the filler (e. g. copper) may be a relatively high friction material.

DISCUSSION. (A) Effect of Load. Variation of the coefficient of friction with applied load for dry, water-, and oil-lubricated conditions is shown in Figs. 3, 4, and 5 respectively, with the Teflon-type materials being represented by solid lines and the other plastics by dotted lines. Although the graphs show only data obtained at a speed of 367 SFM, the same general pattern was found at the other four test speeds. The speed of 367 SFM was selected for the graphs because it best represented the lower coefficients of friction when all the plastics were considered. Actually, as mentioned before, the Teflon-type materials had their lowest "dry" friction values at 8 SFM, averaging about 0.03 to 0.04 less than the values obtained at 367 SFM.

The coefficients of friction generally decrease with increasing load, the effect being most noticeable between 1 and 3 pounds. Among the theories advanced to explain the phenomenon, perhaps the most acceptable is that involving elastic and plastic deformation of the plastics.[7] At high loads where deformation has been shown to be plastic, the real area of contact, A, is proportion to the applied load, W. As the loads become lighter, there is a gradual change from plastic to elastic deformation, and A becomes proportional to about $W^{2/3}$ under idealized elastic conditions. Now, the coefficient of friction, μ, is normally expressed by the equation:

$$\mu = F/W$$

where, $F = sA$,; F = friction force; s = shear strength of plastic; W = applied load; A = real area of contact. For plastic deformation at heavier loads, F will increase in direct proportion to W, but under elastic conditions at light loads, F will not change in direct proportion to increases in W. Thus, at light loads the coefficient of friction will decrease with an increase in applied load.

(B) Effect of Speed. Variation of friction with speed is shown in Figs. 6, 7, and 8. Under dry lubricated conditions (Fig. 6) there are two general effects. One effect is that the friction may decrease as speed is increased from 8 SFM to 367 SFM. The curves for polystyrene and cellulose acetate butyrate demonstrate this effect. Secondly, friction may become greater as speed is increased up to a certain

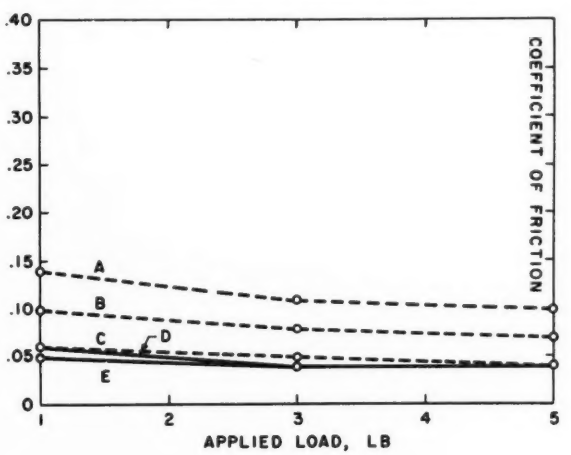

Fig. 5. Effect of load at 367 surface ft/min. using oil lubrication. Key: A, Methacrylate; B, Ethyl Cellulose; C, Polyethylene; D, Teflon Copper (1:1); E, Teflon Asbestos (3:1).

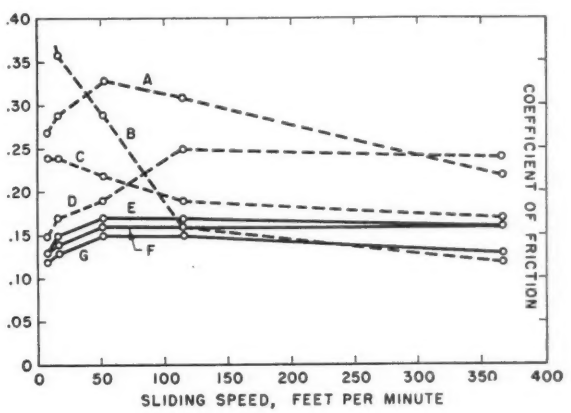

Fig. 6. Effect of speed at 5-lb. load using no lubrication. Key: A, Nylatron-G; B, Polystyrene; C, Cellulose Acetate Butyrate; D, Nylon; E, Teflon MoS_2 (3:1); F, Teflon-Copper (1:1); G, Rulon.

point, then may lessen or level off at speeds beyond that point. Examples of this effect are the curves of nylon, Nylatron G, and the Teflon-type materials.

One possible explanation for the initial rise and subsequent fall of friction with increasing speed might be centered in the combined effects of "stick-slip" and shearing. Stick-slip consists of a combination of static (stick) and kinetic (slip) friction, with static friction being the larger of the two. At very low speeds, the static portion of stick-slip is much in evidence with a correspondingly significant effect in producing large friction values. Also, while speed is being increased in the low speed range, the amount of shearing within the plastic is increasing. Thus, the increase in friction in the low speed range can be a result of increased shearing, with stick-slip contributing its share to the total friction value. Now, as speeds increase above the low speed range, the static friction component becomes more nearly equal to the average kinetic friction component, and this has a reduced effect on the total friction value. At the same time, shearing occurs nearer the surface of the plastic. The result is a leveling off or a decrease in friction at high speeds.

For those plastics that show only a drop in friction as speed increases, the effect of increased shearing at low speeds may be negligible in comparison with the total friction developed by stick-slip. Therefore, high friction at low speed would be primarily due to stick-slip, with the accent on "stick." As speeds increased, "slip" would take over to reduce friction as explained in the directly preceding paragraph.

(C) Effect of Lubrication. The effect of lubrication on the frictional properties of plastics may be found by comparing Figs. 3, 4, and 5, 6, 7 and 8. Changing from dry lubrication to water or oil results in a lowering of friction values as well as a minimizing of the effect of other variables such as load and speed. It might also be expected to narrow the spread between the lowest and highest friction values, and this is entirely true of oil lubrication. However, a reversal of the trend is demonstrated in Figs. 3 and 4 where the spread in water lubrication friction values among the Teflon-type plastics is greater than that obtained under dry lubrication. The reversal is exhibited only by Teflon-copper (1:1) which did not respond to water lubrication to the same extent as the other Teflon-type plastics.

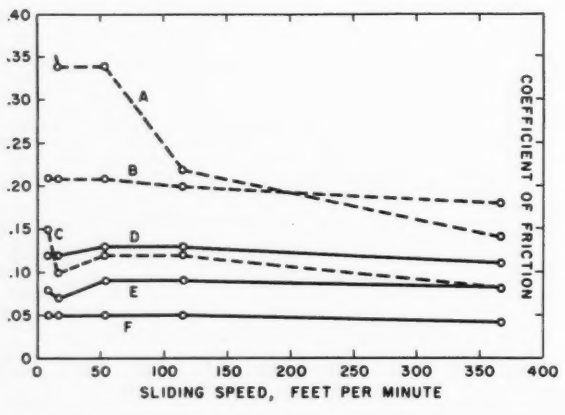

Fig. 7. Effect of speed at 5-lb. load using water lubrication. Key: A, Methacrylate; B, Nylatron-G; C, Polyethylene; D, Teflon-Copper (1:1); E, Teflon; F, Teflon MoS_2 (3:1).

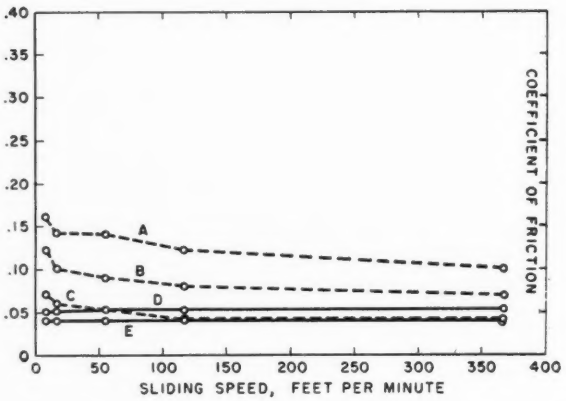

Fig. 8. Effect of speed at 5-lb. load using oil lubrication. Key: A, Methacrylate; B, Cellulose Acetate Butyrate; C, Polyethylene; D, Teflon-Graphite (1:1); E, Rulon.

Although all of the supporting data are not included in this writing, other water lubrication effects were noted. For example, in the non-Teflon type materials, Kel-F and methacrylate are apparently unaffected by water lubrication while polystyrene and cellulose acetate butyrate are affected only at low speeds and high speeds respectively. Cellulose acetate and polyethylene respond especially well to water lubrication.

All the plastics have low friction coefficients when lubricated with oil. Perhaps the most outstanding performance here is that of polyethylene whose friction values drop to a point comparable to those of the Teflon-type plastics. The friction values of cellulose acetate are no lower with oil lubrication than they are with water lubrication at high speeds. In many instances there is no difference in the friction values of filled Teflons when run under water or oil-lubricated conditions. Wetting again appears to be an important factor.

The tests conducted with oil plus butyl stearate lubrication produced only minor variations from the oil lubrication data. Therefore, there will be no special discussion on this phase of the work, except to mention that the virtual absence of any effect of the butyl stearate may correlate very well with the present chemical theories of the action of oiliness additives as expressed by Bowden and Tabor.[1]

Although the testing machine used in this work was designed with the object of eliminating hydrodynamic effects of lubricants, the possibility of the presence of these effects in this investigation is not ruled out.

CONCLUSIONS. Many of the plastics tested show evidence of exceptionally good frictional characteristics even when run dry. Outstanding in this

Table I. List of materials.

	Density	Hardness (Rockwell)
Teflon	2.10	25 (R)
Rulon	2.14	10.5 (R)
Teflon-MoS_2 (1:1)	2.07	63 (R)
Teflon-MoS_2 (3:1)	2.41	33 (R)
Teflon-Graphite (1:1)	2.88	42 (R)
Teflon-Graphite (3:1)	2.12	52 (R)
Teflon-Asbestos (3:1)	2.11	60.5 (R)
Teflon-Copper (1:1)	3.26	54 (R)
Cellulose Acetate	1.24	78 (R)
Cellulose Acetate Butyrate	1.13	72 (R)
Ethyl Cellulose	1.21	69 (R)
Kel-F	2.06	63 (M)
Methacrylate	1.16	89 (M)
Nylatron G	1.14	84 (M)
Nylon	1.14	85 (M)
Polyethylene	0.89	35 (R)
Polystyrene	1.02	81 (M)

respect are the Teflon-type materials, especially at low speeds. The structural and thermal characteristics of Teflon can be changed by adding fillers without sacrificing low friction values. This is also true of nylon and its graphite-filled relative, Nylatron G.

Other comparatively low friction values were obtained with polystyrene run dry at high speed, cellulose acetate and polyethylene with water lubrication, cellulose acetate butyrate with water lubrication at high speed, and polyethylene with oil lubrication.

The friction coefficients usually decrease as the applied load is increased. The most likely explanation for this effect involves elastic deformation at light loads, and plastic deformation at high loads, as discussed in this report.

Increased speed, especially under dry conditions, results in either a drop in friction, or an initial rise and then a drop. For this, a theoretical explanation is advanced which involves combined effects of stick-slip and shearing at low speeds, and effect of shearing only at high speeds.

In general, friction values drop as lubrication is changed from dry to water to oil. Two exceptions are Kel-F and methacrylate, which are unchanged by switching from dry to water lubrication. Another example is cellulose acetate, whose friction coefficients are identical when using either water or oil lubrication at high speeds. Under some conditions, certain Teflon-type plastics are also unaffected by the change from water to oil lubrication,

Table II. Spread in coefficients of friction under dry and oil-lubricated conditions. Applied loads: 1-5 lbs.; sliding speeds: 8-367 ft/min.

Plastic	Lubrication:None		Lubrication:Oil	
	Maximum	Minimum	Maximum	Minimum
Teflon	0.21	0.09	0.06	0.04
Rulon	0.19	0.12	0.06	0.04
Teflon-MoS_2 (1:1)	0.21	0.13	0.06	0.04
Teflon-MoS_2 (3:1)	0.22	0.13	0.06	0.05
Teflon-Graphite (1:1)	0.19	0.12	0.07	0.05
Teflon-Graphite (3:1)	0.19	0.12	0.06	0.04
Teflon-Asbestos (3:1)	0.21	0.14	0.05	0.04
Teflon-Copper (1:1)	0.20	0.13	0.07	0.04
Cellulose Acetate	0.53	0.18	0.16	0.08
Cellulose Acetate Butyrate	0.32	0.17	0.14	0.07
Ethyl Cellulose	0.57	0.22	0.16	0.07
Kel-F	0.25	0.17	0.15	0.08
Methacrylate	0.47	0.16	0.19	0.10
Nylatron G	0.33	0.22	0.13	0.08
Nylon	0.33	0.15	0.14	0.09
Polyethylene	0.40	0.17	0.09	0.04
Polystyrene	0.45	0.12	0.13	0.06

a result which would not be unexpected when considering the extremely low friction coefficients of these materials.

The addition of an oiliness agent such as butyl stearate to the test oil makes little or no difference in the frictional characteristics of the plastics.

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ACKNOWLEDGEMENT. The authors wish to express their thanks to Mr. F. R. Austin who carried out most of the experimental work covered in this paper.

COMMENTARY, by Dr. K. V. Shooter (Chester Beatty Research Institute, London). The wide range of experimental conditions that Milz & Sargent have investigated and the number of plastics that they have used has produced an extremely useful body of data. In this short comment I shall restrict myself to two general points.

Firstly, the geometry of the sliding members used by Dr. Tabor and myself differs from that described by Milz & Sargent. We used a hemisphere sliding on a flat and with this system found that increases in the coefficient of friction which are attributed to a changeover from a plastic to a more nearly elastic deformation only occur at loads of less than 100 gms (1/5 lbs) and in many cases only below loads of 1 gm. It is possible that with the system used by Milz & Sargent the changeover to elastic deformation occurs at higher loads but I feel that this cannot be the explanation for the continued fall of the friction with ethyl cellulose at 3 and 5 lbs load. The odd result obtained with

nylatron suggests that some other explanation is required.

A more important point that Milz & Sargent have not discussed in their paper is the effect of frictional heating. Even at the slowest sliding speed that has been used the frictional heating at the areas of intimate contact will be considerable, temperatures close to the softening point or transition point of the plastics may be easily attained. At the higher sliding speeds the frictional heating will be correspondingly greater and the effects of this heating may extend below the surface layers of the plastic. Because of this heating effect the properties of the plastic which determine the friction, the deformability and shear strength, will be those that obtain at temperatures well above room temperature. For example the marked fall in the friction of polystyrene at sliding speeds up to 100 ft/min. may be due to a progressive fall in the shear strength of the plastic as the frictional heating increases, until at 100 ft/sec. the surface layers may be melted. At this point further increase in the speed of sliding would not be expected to produce much change in the friction thus explaining the small decrease in friction observed between sliding speeds of 100 and 350 ft./min. The nylon type plastics, on the other hand (cf. Fig. 6), show an increase of friction as the speed of sliding is increased. A possible explanation for this is that frictional heating increases the deformability of the nylon giving a larger area of contact during sliding, and if the shear strength of the plastic does not decrease in proportion then the measured friction will increase. These suggestions are clearly tentative and could only be validated by investigation of the effect of temperature on the physical properties involved. The issue is certainly complicated since the explanation of the changes in friction suggested by Milz & Sargent must also be born in mind. With plastics, the area of contact under a given load will in general increase with time of application of the load. As the speed of sliding is increased the asperities on the surfaces will be pressed together for shorter and shorter times and hence as the speed of sliding increases the real area of contact may decrease. This, again, will lead to a reduction of the friction.

COMMENTARY, by Dr. W. F. Busse (E. I. duPont de-Nemours & Co., Inc.). The authors have presented data on the coefficient of friction of plastics which should be of considerable interest to lubrication engineers. However, the data raise one question of both practical and scientific interest. This is: Why is the coefficient of friction found by the authors for the unlubricated friction of "Teflon" on steel (0.09 → 0.21) and polyethylene on steel (0.17 → 0.40) higher than data reported by other workers? Shooter & Thomas (Research, Vol. 2, 533, 1949), for example, report values of 0.04 → 0.10 for "Teflon" and 0.15 → 0.2 for polyethylene on steel.

One possibility is that the literature data were taken at very low speeds (0.02 to 2.0 fpm). The present authors' data show the coefficient of friction of unlubricated "Teflon" to decrease markedly as the speed is reduced from 50 to about 10 fpm, so this factor might have some effect.

Another factor that might play a part is the unit load. Y-H Pao of this Laboratory has calculated the areas of the elliptical regions of contact of the crossed cylinders, using the Hertz theory of elastic deformations and nominal values for the moduli of steel and the plastics. In the case of a hard and a soft plastic, he obtained the following results for the area and the average contact pressures:

Plastic	Load	Contact Area			Average Unit Pressure psi
		Major Axis 10^{-2} in	Minor Axis 10^{-2} in	Area 10^{-4} in^2	
"Teflon"	1 lb	7.2	2.4	14.	700
"Teflon"	5 lb	12	4.08	38.	1,300
Polystyrene	1 lb	2.54	.87	1.7	6,000
Polystyrene	5 lb	4.36	1.5	5.1	10,000

Thompson, Turrell & Sandt (Society Plastics Engineers, 11th Annual National Technical Conference, January 1955, Vol. I, p. 484) have shown that, when using a flat steel ring sliding on a ring of "Teflon" as in a thrust bearing, the coefficient of friction of steel on unlubricated "Teflon" decreased from about 0.03 at 400 psi to 0.016 at 3,500 psi, and this value was constant from 25 to 300 C.

Thompson, Turrell & Sandt also found the coefficient of friction to be sensitive to surface finish of the steel, being 0.035 for surface finished with No. 600 Carborundum, and only 0.016 for a micropolished surface.

Since the present authors used micropolished steel cylinders, this leaves open the question of why they got dry friction values significantly higher than previously reported values.

The changes in friction on going from dry to water or oil lubrication suggest that liquid film lubrication may play a part in the results when oil or water are present, particularly with the softer plastics. When rubber or another soft polymer slides on a smooth metal shaft, film lubrication can be obtained with only moderate speeds, and with pressures of up to 100 psi or more, even with a liquid of as low a viscosity as water, (Busse & Denton, "Water Lubricated Soft Rubber Bearings," Trans. Am. Soc. Mech. Eng., Iron & Steel Division, 54-2, 1932; Beatty & Cornell: "Laboratory Testing of Rubber Bearings," India Rubber World, November & December, 1949). It may be that the elastic material forms an adjustable wedge where the lubricant enters, allowing it to exert the maximum force in keeping the two surfaces apart, analogous to the action of a Kingsbury thrust bearing. Thus it seems probable that hydrodynamic effects do play a part when soft plastics are tested with water or oil as a lubricant in the equipment used by Milz & Sargent.

AUTHORS' CLOSURE. We wish to thank Dr. Shooter and Dr. Busse for their comments. In advancing the explanations given in the paper, it was realized that many factors probably played a part in bringing about the variations in friction. Although no attempt has been made, as yet, to measure temperatures at the point of contact between specimens during a test run, we agree with Dr. Shooter that frictional heating could contribute to the noted results.

In respect to the concern over the fact that some of our coefficients of friction do not exactly coincide with previously reported figures, we wish to re-emphasize the points mentioned in the paper. The objective of this work was not only that of comparing frictional characteristics of the plastics, but also to demonstrate that changes in testing procedure can cause large variations in the coefficient of friction of a material. Therefore, when comparing frictional characteristics of materials, it is extremely important to make certain that the data are, or have been, obtained under comparable testing conditions.

The possibility of hydrodynamic effects being present is well recognized although, as mentioned in the paper, the crossed cylinders apparatus is designed to minimize such effects. Actually, the level of coefficients of friction which were obtained indicate that we were probably operating in a combined hydrodynamic and boundary lubricated region.

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Hermetically Sealed Refrigeration Compressor Lubrication

by K. Borah*

The introduction of hermetically sealed refrigeration compressors solved many mechanical problems of the refrigerator, but created a need for a lubricant having an unusual combination of properties, including high stability, prevention of copper plating, solubility for the refrigerant, good flow at low temperature, resistance to moisture, non-foaming, and good lubrication properties.

The utilization of modern hermetically sealed refrigeration compressors has created one of the most highly specialized applications of petroleum lubricants in modern machine design. Few known mechanical devices in wide spread use place so great a burden on a lubricant, and probably in no other application does the lubricant become so intimately a part of the device or more actually a material of construction.

In the early days of mechanical refrigeration the lubrication of compressors was a fairly straightforward application. The compressors in commercial installations and even the early domestic units were conventional reciprocating types, and the lubrication of such machines posed no serious problems. The lubricant was generally applied to the smaller units by splash from the crankcase and to the larger machines by forced circulation to the bearings and direct injection to the cylinders from exterior lubricators. Stability of the lubricant was not a critical factor as the operating temperature of the compressors was usually quite moderate and the oil was changed at reasonable intervals. In the past the most important consideration in the selection of a lubricant for refrigerator service was the ability of the oil to flow at low temperature. Most refrigerants in common use in those days were not miscible in lubricating oils and, therefore, it was essential that any oil reaching the cold side of the system remain fluid in order that it could be drained off or returned to the crankcase. It was that factor in performance that so firmly established low pour point as an essential oil property in the minds of those in the refrigeration industry.

While the use of compression systems for refrigeration continued to grow in commercial applications, it was actually the introduction of mechanical refrigeration for home use that launched the industry on a new era of development. Not only has the production of domestic units grown into a gigantic industry on its own, but many of the principles of design developed for household refrigeration are now in use in the commercial field. The early adaptation of mechanical refrigeration to domestic use consisted primarily of reducing the physical dimensions of the larger commercial equipment to roughly those of an old fashion ice box. Although many difficulties were encountered in the first attempts at fitting all the necessary apparatus into so small a space, the results were highly successful. This is indicated by the fact that even as far back as 30 years ago approximately 1,000 machines per month were being produced and sold at prices roughly double those obtained for present day units.

The early machines, as compared with present day units, did, of course, have certain objectionable features. The three major problems with the mechanical features of the units were durability, space requirements, and noise. The units were by no means trouble-free and required considerable servicing. The compressors were for the most part rather bulky reciprocating types generally belt driven by separately mounted electric motors. In most cases fans for cooling the condensers were also included in the equipment. This equipment required a large portion of the space available and allowed only limited space for food storage. The operation of the compressor, motor, fan, and belt were indeed noisy and in many cases the refrigerator could be heard throughout the house.

The methods by which the three major difficulties were overcome involved many problems in design, materials, and mechanics, but the end result which made the greatest contribution to a solution of all three problems was the development of the hermetically sealed compressor. The reliability of the sealed compressor is now well established as hundreds of thousands of such units have been in service for periods exceeding 10 years without requiring even minor adjustments. The units require only a fraction of the space needed for the older machines, and the storage space available in modern refrigerators is almost double that of the older models occupying the same amount of floor space. The noise level has been reduced to the extent that a unit in operation can hardly be heard even in the same room.

The development of the compact sealed compressor not only solved many of the problems of the home refrigerator but also opened up entirely new fields in both the domestic and commercial use of mechanical refrigeration. Home freezers, packaged air conditioners, refrigerated food display cabinets, soft drink dispensers, and dehumidifiers are just a few of the items now being equipped with this type of unit.

The hermetically sealed unit consists of a complete compressor and a direct coupled electric mo-

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tor, both of which are enclosed within a sealed welded steel case. The case serves as a housing for the equipment and also functions as a reservoir for the lubricating oil and as a suction vessel for the refrigerant. The cooling fan has been eliminated on many of the smaller units and all moving parts of the refrigerator are contained within the case. The lubricant is applied to the equipment by conventional methods involving the use of oil pumps which force the oil through drilled or spiral grooved passages to the various parts of the units. In some units where the motors are mounted in a vertical position, the oil is actually flowed over the windings to improve cooling. The quantity of lubricant is generally quite small and in some cases with household units the normal filling is less than one pint.

While this discussion deals primarily with the hermetically sealed compressor originating with the household refrigerators, we should not overlook the fact that on many of the larger commercial units, sealed systems are now being employed and the lubrication requirements are essentially the same as for the smaller machines. Units with internal motors are being used with centrally located air conditioning systems of several tons capacity, and even where external drives are used, the compressor crankcases are generally sealed and serve both as oil reservoirs and suction vessels.

Another important factor in the development of modern refrigeration systems is the development of more suitable refrigerants. The most outstanding of these materials are the halongenated hydrocarbon compounds, known by the trade name "Freon". One of these materials, "Freon 12" (CCl_2F_2) dichlorodifluoromethane, is widely used in equipment where sealed compressors are installed. This material differs rather widely from most of the early refrigerants, particularly from the standpoint of lubrication, as it is highly soluble in mineral lubricating oils.

A modern mechanical refrigeration system consists primarily of four major components: a sealed motor driven compressor, an air-cooled tubular condenser, an expansion device for releasing the refrigerant, and an evaporation chamber. The cycle involves compressing the gaseous refrigerant and discharging it to the condenser where it is liquefied and then released through a float controlled valve or a capillary tube and allowed to expand in the evaporator. It is, of course, in the evaporator that heat absorption occurs and the intended work of the system is performed. The refrigerant, after evaporating, is then returned to the compressor case and to the compressor. The frequency with which the compression cycle occurs is, of course, dependent on the design of the units and the type of service, and varies from infrequent operation of short duration to continuous operation in certain types of commercial installations.

As further progress was made in the development of the sealed compressors the units were made more and more compact, the operating speeds increased, and higher and higher operating temperatures encountered. In many instances the severity of the operating conditions under which the lubricating oil must perform began to exceed those of older equipment, and it became apparent that the quality of the lubricant was of vital importance to the operation of the system. It was at this stage of development that manufacturers began to recognize the fact that many of the earlier oils that had served the industry so well were no longer satisfactory and that properties other than pour point and viscosity were significant in the performance of the lubricant.

One outstanding factor in the lubrication of hermetically sealed compressors that make their lubrication unique and emphasizes the importance of oil quality is the fact that the oil is seldom if ever changed and no make-up oil is added during the life of the machine.

Much of the need for an improved lubricant was met by the development of more highly refined versions of the early "ice machine" oils, and these naphthenic base, low pour point, low viscosity index, materials are still being used successfully in many types of sealed compressors. In some cases naphthenic oils refined to the degree of medicinal white oils have been found satisfactory in compressor applications.

With the continued extensive use of oil soluble refrigerants, the high viscosity index paraffinic base oils were also introduced for refrigerator lubrication. The advantages of high viscosity index particularly in applications where temperature varies widely and the stability and susceptibility to fortification by additives are well known. Many of the lubrication requirements of sealed compressors make these characteristics highly desirable.

Experience obtained in the lubrication of present day equipment has well established the fact that there are many properties in an oil that are important in this service, and it is difficult to rate the various properties in the order of their importance as variations in machine design and operation place a varying degree of emphasis on the different properties. However, the properties discussed here are all important to good lubrication and each one should be given careful consideration when the selection of a lubricant is made.

Stability. The primary consideration regarding oil stability in sealed compressor service is the tendency of an oil to form sludge, varnish, or other undesirable deposits. The clearances in such equipment are usually quite small, in some cases on the order of 0.0001", and any appreciable deposition will cause excessive drag if not actual sticking of the unit.

Stability of an oil in refrigeration service is a critical factor due to three principal reasons.

1. The oil is subjected to high operating temperatures.

2. A small quantity of oil is exposed to relatively large surface areas.

3. The oil is in intimate contact with various materials such as refrigerants, metals, insulating materials, etc., that may in service combine with the oil in such a way as to adversely affect its stability.

The operating temperatures in sealed compressors are frequently higher than those encountered in other types of equipment, as the compressor heads or valve assemblies may rise above 300 F. and the bulk oil temperature may at certain times during the compression cycle go well above 200 F. The oil as circulated passes over such metallic surfaces as steel and copper both of which can act as catalysts and affect the rate of deterioration of an oil. Other materials of the systems such as refrigerants, even the halogenated compounds, can by hydrolysis, where small amounts of water or air are present, form materials harmful to the oil and it has also been fairly well established that certain decomposition products of an unstable oil can have a deleterious effect on a refrigerant.

It is well known that extraneous materials such as water and oxygen, when present even in trace quantities in a refrigeration unit, can greatly accelerate decomposition of both the oil and the refrigerant and every effort is usually made to exclude these materials from the system. However, it has also been shown by laboratory tests that even when very precise methods are used to exclude these materials, harmful deposits can still be formed.

Careful analyses of such deposits have almost always revealed the presence of chloride and/or fluoride ions which indicates that some chemical decomposition of the refrigerant takes place. The extent to which one material influences the decomposition of the other, and the exact sequence of events, is not fully known, but it has been determined that the quantity of deposits formed varies considerably with different oils.

Much work has been done both experimentally and by fundamental studies of the oil stability problem in refrigerator systems. While the progress of this work has been slow, due to the complexity of the problem, certain fundamental facts of importance regarding the performance of lubricants have been established. It is now recognized that the more highly refined oils are in many cases more satisfactory for this service. Certain additives when used in combination with oils of proper composition and refinement are effective in improving oil stability in refrigeration service and are also being used.

Extensive laboratory testing has rather clearly indicated that no single one of the factors mentioned above necessarily will make an oil suitable for this service but, rather, that it is a proper combination of the various factors. The method and degree of refining of an oil of improper hydrocarbon composition may not make it a good refrigerator lubricant. While the stability of paraffinic type oils is well known, they will not necessarily give satisfactory performance if they are not properly refined and fortified. Numerous additives have been known to improve the stability of oils under a variety of conditions, but many such materials may be completely ineffective in refrigerator service and in some cases may react with the refrigerants or other components of the system in such a way as to be actually harmful.

The lack of more convenient test procedures makes the study of oil stability in refrigeration systems both difficult and time consuming. While certain laboratory procedures have been useful as initial screening steps, the final and complete approval of an oil by most refrigerator manufacturers requires testing in actual units under severe operating conditions for periods of two to three years duration. In view of the numerous variables involved in such a system, and the wide differences in operating conditions of various installations, it appears doubtful that suitable short time laboratory test procedures will be developed for evaluating oils for this service.

Low Temperature Properties. The low temperature performance characteristics of lubricating oils in present day refrigeration equipment has probably been the cause of more controversy and misunderstanding than any other single factor in compressor lubrication.

Only the compressors and motors require lubrication, but with the use of refrigerants of high oil solubility at least some of the oil passes through with the compressed gas; therefore, consideration must also be given to the performance of the oil in other parts of the system. In the earlier days with the use of such refrigerants as ammonia, sulfur dioxide and carbon dioxide that were not miscible with oils, it was essential that the lubricant, in an undiluted form, remain in a fluid state at the lowest temperature encountered in a system. The oil that reached the cold parts of a system was forced there by the flow of the refrigerant and if upon reaching the evaporator the oil tended to solidify or become too viscous to be moved on by the refrigerant, the evaporator walls would become coated and serious interference with heat transfer and refrigerant flow would occur.

The ASTM pour point procedure was a satisfactory method for determining the suitability of an oil's low temperature performance with refrigerants of low or negative oil miscibility. However, with "Freon 12" or any highly oil soluble refrigerant, pour point values are of limited significance. It must be understood that pour point is a measurement of the temperature at which an undiluted oil changes from a liquid to a semi-fluid state. With the use of oil soluble refrigerants it is the low temperature performance of the solution that is the important factor.

Normally there are two factors involved in the low temperature performance of an oil. First, viscosity is important with neat oils but on the cold side of refrigerator systems the oil content of the solution will probably not exceed 3 or 4 percent and therefore viscosity at this point is of limited importance. Second, the wax components of the oil at low temperatures is a factor of paramount importance even when the oil is diluted by the refrigerant. In consideration of the effect of wax components on performance, it must be recognized that several types of waxes can be present in petroleum oils and that the various types can be removed from some oils with considerable degree of selectivity by the refinery dewaxing process. The only types of

waxes that cause difficulty in a refrigeration system are those which precipitate from the oil and refrigerant and remain in an undissolved state to cause loss of heat transfer or plugging of the expansion mechanism. Much of the wax found in oils is entirely soluble in some "Freons" and will not precipitate at any temperature encountered in the system; therefore, it is not a matter of pour point of the neat oil but, rather, the removal by selective dewaxing of those portions of the wax components that do not remain dissolved in a refrigerant.

Oils with pour points as much as 60 to 70 degrees above the evaporator temperature have been known to perform entirely satisfactorily. Oils of plus 10 F. pour point have been used for years in thousands of refrigeration units with no evidence of difficulty.

Unfortunately none of the accepted laboratory procedures can be relied upon to determine accurately the low temperature suitability of an oil in refrigeration service. Pour point and cloud point tests deal only with neat oils. Floc point, a test used by some refrigerator manufacturers, is somewhat more useful in that it involves the testing of a sample of oil in solution in "Freon". However, this test does not determine the nature of the wax that causes the clouding or floc, and oils of low floc points have been known to separate small amounts of hard waxes that would not redissolve.

Due to the shortcomings of standard test procedures, some manufacturers study oils by testing them in service in units with transparent traps installed in the evaporator where any precipitation of wax can be visually observed.

Viscosity. Viscosity, as is true in most applications, is an important property in the selection of a lube oil for refrigeration service. In sealed systems the oil must not only lubricate all of the various parts of the compressor but must also aid in sealing in both rotary and reciprocating units. Many of the smaller reciprocating compressors do not employ piston rings and, therefore, the oil is relied upon to perform the entire sealing job. The lubrication requirements of such machines are usually quite critical for, as is the case with many mechanisms where conservation of space is a prime requisite, the unit loads are often quite high.

The matter of oil viscosity in systems operating with "Freon 12" becomes rather complex due to dilution by the refrigerant. At no time during the refrigeration cycle is the oil entirely free of refrigerant and the degree of dilution varies rather widely. The greatest amount of dilution in the bulk oil will generally occur at the start of the refrigeration cycle when the pressure in the case is the highest and the temperature of the oil is the lowest. As the compressor operates the pressure is rapidly reduced and the bulk oil becomes fairly free of refrigerant, but at points in the machine where the actual compression takes place the dilution remains quite high. Therefore, dilution, load, clearance, speed, and temperature must all be taken into account when considering viscosity, and a proper selection of a lubricant can not be made without complete information on the particular machine involved.

Some manufacturers determine the proper oil viscosity for their units by measurements of the volumetric efficiency of the system. This procedure will, of course, take into account all of the various factors encountered in the unit, many of which probably cannot be accurately duplicated by any other method.

Viscosity Index. Oils of high viscosity index are desirable in refrigerator service for essentially the same reasons that make this property important in other applications. It is highly important that the oil remain sufficiently viscous at the highest temperature encountered to seal properly and lubricate, and it is, of course, undesirable to have the oil too viscous at low temperatures.

The important consideration in selecting an oil for any application is the required viscosity at the highest temperature encountered in the equipment. As an example, if the normal operating temperature of the compressor is 180 F., and it has been determined that a viscosity of 80 Saybolt seconds at the operating temperature is required to maintain adequate oil film for proper lubrication and sealing, it can then be considered that this is the maximum viscosity requirement for the unit. However, since the viscosity of oils changes with temperature, the oil will, at any point below the normal operating temperature, be more viscous than is necessary for the application and at higher than normal operating temperature may fail to provide sufficient seal. In this regard, viscosity index becomes important as a low viscosity index oil with a viscosity of 80 seconds at 180 degrees will have a viscosity at 100 degrees of over 600 seconds, whereas a suitable high viscosity index oil will be in the range of 400 seconds at 100 F. At a room temperature of say 70 F., which may be approximately the temperature at which the compressor is expected to start, the correct grade of LVI oil will be about 2200 seconds and the HVI oil only about 1200 seconds.

In consideration of the energy required to start the compressor, and the time and energy required to start the flow of oil to the machine, this difference in viscosity becomes important particularly in small units in the fractional horsepower range. If for reasons such as high ambient temperature or heavy loading the operating temperature should be appreciably increased, an oil that tends to change widely with the temperature may decrease in viscosity to the point of failure to seal or lubricate. Some observers have indicated that high viscosity index oils may give greater volumetric efficiency than the corresponding grade of LVI oils when the temperature is increased 30 to 40 degrees above the normal operating range.

While it is recognized that the oil in a compressor is in a diluted form, viscosity index is still an important factor as blends of oils and refrigerants respond to temperature changes in the same manner as neat oils. A viscosity index of 90 minimum is specified by some manufacturers.

Dielectric Strength and Moisture. The electrical insulating quality of a refrigerator oil is im-

portant in hermetically sealed compressors as the oil is in direct contact with the motor.

Most properly refined refrigerator lubricants have insulating ability far beyond the requirements of the application; however, even minute quantities of water can be deleterious to both its insulating ability and, as previously mentioned, the stability of the refrigeration system. It is common practice for refrigerator manufacturers to use the standard A.S.T.M. dielectric strength test for examining lubricants and, although it is in this application primarily a test for moisture, it appears to be a suitable procedure. A minimum dielectric strength of 25 kv is generally required by manufacturers.

It should be remembered that freedom from water is extremely important in refrigeration systems, even when the motor is mounted externally, and some suitable method of testing the oil for moisture should always be employed.

Volatility. Volatility of the lubricant is important in refrigerator application both in service and during assembly of the unit. In service the high local temperatures encountered in the compressor may tend to vaporize any light components present in an oil and cause both a change in its physical properties and excessive carry-over into the high pressure side of the system.

In the factory assembly of the machines the compressor unit is usually evacuated under both high vacuum and high temperature as a means of removing oxygen and moisture, and this procedure also tends to vaporize the more volatile fractions of an oil.

It should be remembered that oils of different hydrocarbon composition vary considerably as to volatility. The paraffinic oils are appreciably less volatile than naphthenic oils of the same viscosity. Flash point as related to volatility indicates this quite clearly. The flash point of a paraffinic oil is usually about 50 to 60 degrees higher than for the corresponding grade of naphthenic oil. The flash point of HVI oils in the viscosity range suitable for most refrigerators will be on the order of 400 to 500 degrees. Flash point alone is not necessarily a good measure of volatility and some manufacturers have been forced to design their own tests for examining this property.

Resistance to Foaming. The operating condition of the sealed compressor system are conducive to foaming of the lubricating oil.

The optimum conditions for foam formation exist at the start of the cycle, when a rapid pressure drop occurs in the crankcase of the gaseous refrigerant absorbed in the oil during the idle period is released. If the oil has a tendency to form a high foam layer, an excessive quantity of the oil is likely to become entrained in the gas flow to the compressor suction and thereby be carried into the high pressure side of the system.

Several refrigerator manufacturers have special foaming tests of their own design and observe this property of an oil very carefully. Both the amount of foam formed and the time required for the foam to collapse are important factors, and in no case should an oil that tends to form a stable foam layer be considered suitable for refrigerator service. A suitable oil should give non-foaming results through all three sequences of the A.S.T.M. D892-46T foaming test.

Copper Plating. It has been noted with some refrigeration equipment that copper from the various lines and perhaps from the windings of the motor may be taken into solution by the oil and/or refrigerant. The copper then tends to plate out on various other metals in the equipment such as the polished steel faces of bearing journals and compressor rotors, etc., and, since the clearances are usually quite small, this can become a serious problem. This phenomenon can not be entirely attributed to the oil as it has been known to occur in the presence of the refrigerant alone. However, it has also been observed that the reaction is more pronounced in the presence of some oils when in solution with the refrigerant.

The role played by the oil in copper plating is not fully understood, but experimental work has indicated that it is closely related to the problem of oil stability and it has been found that less copper plating will occur when more stable oils are used. The same type of oil found most suitable from the standpoint of the formation of other harmful deposits has also been found to reduce copper plating to a satisfactory minimum.

The severity of copper plating has been greatly reduced with the use of halogenated refrigerants but the problem still warrants vigilance on the part of the manufacturers, particularly regarding the purity of the refrigerant and the stability and purity of the lubricant.

Summary. To summarize, the important characteristics of a refrigerator oil are as follows:

(1) It should have the best possible stability against deterioration.

(2) It should be free of all waxes that will not remain dissolved in the refrigerant at the lowest temperature encountered in the system.

(3) The viscosity should be the lowest that will give adequate sealing and proper lubrication through the range of dilution and temperatures encountered.

(4) Viscosity index is an important factor regardless of whether or not the refrigerant and oil are miscible.

(5) The oil should have high dielectric strength and be free of moisture.

(6) The lightest fraction of the oil should not volatilize at the highest temperature encountered in the compressor.

(7) It should have a minimum tendency towards foaming.

Future Requirements. Even though the increase in the use of mechanical refrigeration in both domestic and commercial applications has been phenomenal in the past 25 years, it now appears that it has by no means reached even a leveling off point.

The wide acceptance of air conditioning for passenger automobiles and the extremely rapid increase

in home air conditioning is an indication of the pace at which this industry is moving. That many homes are to be equipped with a refrigerator, a food freezer, a dehumidifier, an air conditioned automobile, and one or more unit air conditioners, has already become a matter of fact.

It is interesting to note that practically all new uses for mechanical refrigeration continue to emphasize more and more the need for increased refrigeration capacity from mechanical units of smaller physical dimensions. Needless to say this trend has created many problems in the manufacture and operation of the equipment. The development of such equipment has been largely the result of successful effort on the part ot research groups in finding improved materials of construction, and an appreciable part of this effort has been directed towards studies of lubricants and their performance. To date the lubrication requirements of the machines have been successfully handled by petroleum base oils, and lubrication has not as yet been a limiting factor in compressor design. However, in consideration of possible further increases in the severity of operating conditions and changes in compatibility requirements that may come about by the use of new materials of construction and/or new refrigerants, the need for continuing the study of lubricants is evident. The high temperatures often encountered in such equipment as automobile air conditioners and the more compact and higher capacity room coolers certainly indicate that thermal stability, viscosity over the operating range, and volatility will continue to be important factors. Refrigerants which exhibit sharp changes in oil solubility characteristics within the temperature range of the cycle are now being used in many systems and the use of such refrigerants will no doubt continue to focus considerable attention on the low temperature performance of oils.

Previous work has clearly indicated that due to the severity and specialized nature of the application, many of the standard testing procedures in common use for the evaluation of lubricating oils are of only limited value for predicting the performance of an oil in refrigeration service. The need for additional laboratory methods for studies of such factors as compatibility with the refrigerant, thermal stability, volatility, and low temperature performance is apparent. However, experience in actual service has clearly indicated that laboratory evaluation methods, while useful in the development of lubricants, will confirm the fact that a suitable lubricant must be a material designed, manufactured, and applied with the same degree of precision and technical consideration that is given to all other materials that make up the numerous component parts of the equipment.

Section News

BOSTON, July meeting. Election of Officers (see ASLE Directory. Submitted by E. B. Harvey, Jr., former Sec'y.)

LOS ANGELES, October. J. J. O'Connor, Managing Editor of "Power" magazine, presented a paper entitled "Where We Stand Today on Fire-Resistant Turbine Fluids."

1955-1956 Program. **November 8,** J. T. Wilson, The Louis Allis Co., "The Effects of Atmospheric Contaminants Upon Grease & Oil Lubrication Applications." **January 10, 1956,** L. D. Cobb, New Departure Div., GMC, "New Departure Ball Bearing Research Activities." **February 14,** H. P. Henderson, Worthington Corp., "Lubrication Problems of Gas & Air Compressors." **March 13,** J. P. Critchlow, Gulf Oil Corp., "Gear Lubrication in the Steel Industry." **April 10,** M. K. Gebert, National Motor Bearing Co., "Oil Seals — History, Design & Application." **May 8,** D. F. Wilcock, General Electric Co., "The How & Why of Ball Bearing Lubrication." **June,** Field Trip. (Submitted by V. V. Mason & C. A. Barrand, Program & Ass't. Program Chrmn.)

N. CALIFORNIA, October. J. J. O'Connor, Managing Editor of "Power" magazine, presented a paper entitled "Where We Stand

(Continued on p. 351)

TWIN CITIES, Lubrication Engineering Course. Ninety-seven students enrolled in the condensed evening course on "Basic Fundamentals of Lubrication" sponsored by the ASLE Twin Cities Section, and held in February & March at the Dunwoody Industrial Institute. Sessions included: (1) Principles of Lubrication, (2) Properties of Lubricants, (3) Greases for Industrial Use, (4) Application of Lubricants, (5) Hydraulic Oils and Hydraulic Machinery, (6) Bearing Lubrication, (7) Recent Developments in Lubrication, (8) Analysis of Failures, (9) Proper Purchasing, Storage & Handling of Lubricants, and (10) Panel Discussion on Lubrication Factors. (Submitted by B. T. Harding, Membership Committee Chrmn.)

Factors Influencing Friction & Wear With Solid Lubricants

by M. B. Peterson & R. L. Johnson*

An experimental study was made of solid lubricants and the factors that influence their lubricating effectiveness. The research was conducted using a low speed friction and wear apparatus with steel test specimens. In most cases the lubricants were used in the powder form. Most data were obtained with MoS_2 but other solid lubricants were also studied.

Moisture is harmful to lubrication with MoS_2, but may be a necessity for effective lubrication under some conditions with other solid lubricants. Lubrication effectiveness of MoS_2 improves with increasing temperatures which minimize the effects of moisture. Very small amounts of abrasive contaminants in solid lubricants can have considerable effect on wear. Mixtures of MoS_2 and oil must contain at least 10 percent of either MoS_2 or oil to obtain the minimum friction coefficient.

Crystal structure can be a useful guide in the selection of solid lubricants with low shear strength. However, ability of the solid to form an adherent film on the surface to be lubricated is the primary essential for good lubrication by solids.

Increased emphasis has been given to solids as lubricants because of the possibility of their use at high temperatures,[1] at high loads, and in inaccessible machine parts. Since the solids are used in many different ways (in oil, bonded films, greases, dry powder, etc.) the problem of solid lubrication is two fold: to find a good solid and to determine the best way to use it.

Concepts developed for lubrication with liquids and soluble additives are not applicable to the case of solid lubricants. The influence of operating parameters on the effectiveness of lubrication by solids has not been established. Basic generalizations in solid film lubrication could aid in the solution of many practical problems.

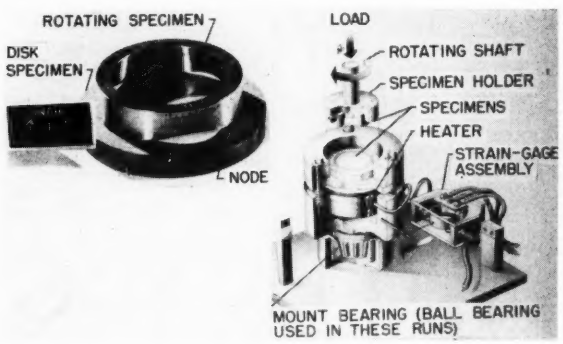

Fig. 1. Detail of basic elements of kinetic-friction apparatus. 1a (left) Photograph of slider and disk; 1b (right) Diagrammatic sketch.

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(This paper was sponsored by the ASLE Technical Committee on Lubrication Fundamentals, and presented at the ASLE 10th Annual Meeting, Chicago, April 14, 1955.)

The theory of solid film lubrication which has been advanced by Bowden and others[2] is as follows: The coefficient of friction for unlubricated metals has been expressed as

$$f = \frac{\text{area of shear} \times \text{shear strength}}{\text{load}}$$

The ratio $L(\text{Load})/a(\text{area}) = \text{constant} = p = \text{flow}$ pressure of the softer of the two mating parts. When a thin film of lubricating material is placed between the surfaces, the friction coefficient can also be expressed

$$f = \frac{as}{L} = \frac{s}{p}$$

where p is the contact pressure and s is the shear strength of the film. If there is some metal contact through the film, the coefficient of friction will be increased in proportion to α, the fraction of the total shear area where metal to metal contact occurs. This may be represented

$$f = \frac{\alpha\, s_{\text{metal}}}{p} + (1-\alpha)\,\frac{s_{\text{film}}}{p}$$

Any factors which could influence this relationship are important in lubrication by solid films and some are considered in the course of the research reported herein.

It is the objective of the research reported herein to obtain some guides to indicate the influence of various factors (such as environment and method of use) on the effectiveness of a typical solid lubricant and also to evaluate a number of possible solid lubricants in order to obtain an understanding of the basis for their lubricating effectiveness.

Friction tests were therefore run on a low speed kinetic friction apparatus. The influence of humidity, contamination, and crystal structure on the lubricating effectiveness of MoS_2 and other solids are reflected in kinetic friction and wear measurements. Some of these data are reported in greater detail in Refs. 3, 4 & 5.

APPARATUS & PROCEDURE. Test Specimens. A 2 inch-diameter ring specimen rotated on edge against the flat surface of a disk specimen (Fig. 1a). The ring is made of nickel-molybdenum steel (SAE 4620), case hardened to Rockwell C-62. Three nodes were ground on the edge of the test

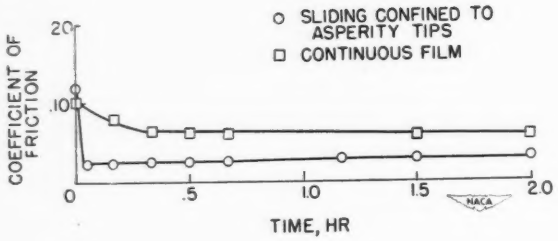

Fig. 2. Effect of sliding time on the coefficient of friction for dusted films where sliding is on the asperity tips and with a continuous film on steel specimens lubricated with MoS_2 powder. Atmosphere, dry air; load, 40 pounds; speed 11 rpm (5.7 feet per minute).

ring that rotated against the disk surface. Each node was in the shape of a cylinder (2 inch radius), and line contact theoretically existed between the three sliding nodes and the disk. The nodes were ground on the rings using a master pattern machined to close tolerances; therefore each ring was essentially identical with all others. Final grinding marks were removed by hand polishing.

The disks used in this investigation were machined from SAE 1020 steel. They were given a vapor blast finish (60 rms) to avoid the directional nature of ground surfaces.

Apparatus. A diagrammatic sketch of the basic parts of the apparatus is shown in Fig. 1b. The ring specimen was mounted in a holder which was held in the spindle of a rotating shaft. The rotating shaft could be raised and lowered or locked in any position. The disk specimen is mounted on the support table and kept from rotating by means of a pin. The support table was mounted on three springs to compensate for any misalinement of specimens.

A load was applied between the specimens by lowering the rotating assembly to deflect the base springs on which the apparatus was mounted (not shown in Fig. 1b). The magnitude of the load was obtained by observing the amount of deflection as indicated by a dial gage. During a test run the desired load was first applied and the spindle then clamped in that position, thus maintaining the load constant throughout the test.

Frictional torque was measured by restraining the motion of the lower assembly through a dynamometer ring on which strain gages had been mounted.

Cleaning Procedure. Prior to each test run, both of the friction specimens were cleaned according to the following procedure: (1) Washed in 50:50 acetone-benzene solution; (2) Scrubbed with repeated applications of moist levigated alumina; (3) Washed in tap water; (4) Washed in distilled water; (5) Washed in 95-percent alcohol; (6) Dried and stored in a desicator.

Test Procedure. A new slider and disk specimen were used for each run after finishing and cleaning as previously described. Approximately 20 grams of the solid lubricant was supplied to the disk surface and a load of 40 pounds was applied. The tests were begun and run to completion without interruption; friction force was continually observed and measurements were recorded approximately every 10 minutes. The amount of wear was determined with a planimeter by measuring areas of wear spots from projected enlarged images of the wear areas on the slider specimens. At the end of each run all specimens were examined microscopically and photographed.

THEORETICAL BASIS FOR EVALUATION OF RESULTS. As was stated previously the equation which has been derived for the friction coefficient in lubrication with solids is

$$f = \alpha \frac{s_m}{p} + (1 - \alpha) \frac{s_{film}}{p}$$

From this equation it is seen that there are several variables which determine the value of the friction coefficient, f.

In the absence of metal-metal contact the ratio s_{film}/p, determines the coefficient of friction. It has been shown by Bridgman[6] and demonstrated by Boyd and Robertson[7] for MoS_2 and other materials, the shear strength versus pressure curves for inorganic compounds are usually of the approximate form $s^2 = mp$; this means that the coefficient of friction will usually decrease with an increase in contact pressure.

The contact pressure, p, may be the flow pressure of the solid lubricant if the film is thick enough; or with decreasing film thickness approach the flow pressure of the base metal. In accordance with the equation[2]

$$f = \alpha \frac{s_m}{p} + (1 - \alpha) \frac{s_f}{p}$$

the coefficient of friction will decrease because of the higher contact pressures that result from thinner lubricant films. This decrease was noted in experiments with bonded films of MoS_2. When a very thick bonded film was applied to both the slider and disk, the coefficient of friction at room humidity was 0.26; with a thinner bonded film on the disk only, the coefficient of friction was 0.09. The "ploughing" effect as described by Bowden and

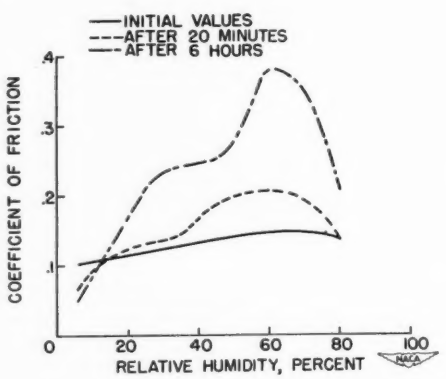

Fig. 3. Effect of various relative humidities on the coefficient of friction of steel specimens lubricated with powdered MoS_2. Load, 40 pounds; speed 11 rpm (5.7 ft./min.).

Tabor (Ref. 2, p. 91) for lubrication by thin metallic films could also contribute to higher friction with greater film thickness. Where the wear track is continually retraced, as in these experiments, the influence of the ploughing effect is believed to be minimized.

There is another value of the contact pressure which could be obtained. Since the asperities in many cases are harder than the bulk metal as a result of work hardening, the contact pressure which would result if the sliding were confined to the tips of the asperities, rather than on a continuous film, would be higher. An increase in pressure would cause a decrease in the coefficient of friction.[2] This change was noted in experiments with solid lubricants and is shown in Fig. 2 for MoS_2. The sliding was confined to the tips of the asperities by lightly dusting the surface with an amount of lubricant that was insufficient to fill the interstices between the asperities. The coefficient of friction for the two conditions differs by a factor of two.

One of the most important variables affecting the coefficient of friction is the amount of metal contact (α) existing during sliding. The amount of metal contact taking place through the lubricant film and the rate of increase of metal contact during sliding is considered to be a measure of the effectiveness of the solid lubricant. Thus in an evaluation of different solid lubricants and also in comparing the different methods of application of any one solid the changes in coefficient of friction resulting from changes in contact pressure and s/p ratio must be separated from those changes which result from a difference in amount of metal contact.

EXPERIMENTAL. Effect of Humidity with MoS_2 Lubricant. In order to determine the effect of humidity the test section was surrounded by a jacket and air of the desired humidity passed through the test-specimen chamber at a rate such that a slight positive pressure existed within the chamber. The relative humidity could be varied by passing different percentages of a stream of dry air through water before allowing it to enter the test section. The relative humidity was measured by means of a hair hygrometer (± 2 percent accuracy) which was repeatedly calibrated against a dew-point potentiometer.

The test specimens were cleaned, placed in the test chamber, and dried for ½ hour by allowing dry air to flow through the enclosing chamber. The MoS_2 was applied and drying continued for another

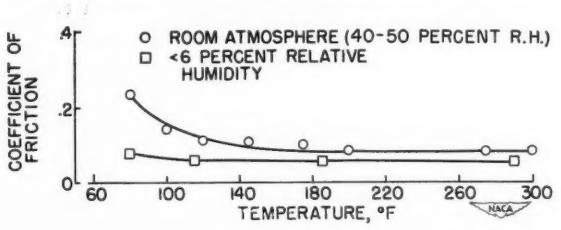

Fig. 4. Effect of temperature on friction of steel surfaces lubricated with purified MoS_2. Load, 40 pounds; sliding velocity, 5.7 ft./min.

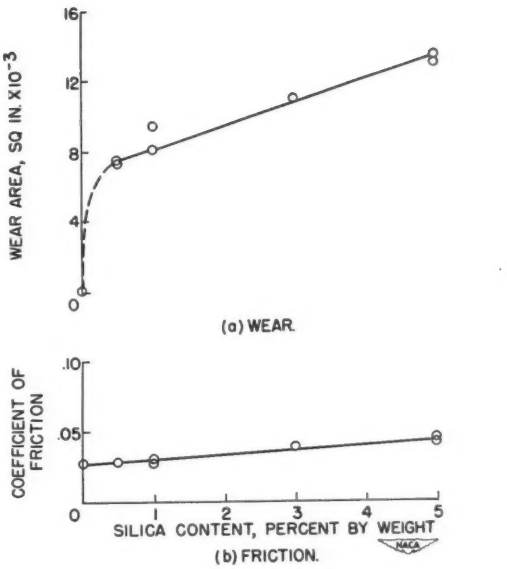

Fig. 5. Effect of additions of silica to a highly purified grade of MoS_2 on wear and friction of steel specimens in dry air. Load, 40 pounds; sliding velocity, 5.7 ft./min.; duration, 6 hrs.

½ hour. The humidity was then adjusted to the desired level and after ½ hour the load was applied and the test begun. Friction force was continually observed. Numerous tests were run at relative humidities of <6, 15, 31, 44, 51, 60, and >70 percent; the data are shown in Fig. 3 where the initial, 20 minute, and 6 hour values of coefficient of friction are plotted against relative humidity. These data show that f increased up to a relative humidity value of approximately 60 percent and then decreased. The reasons for these trends (up to 60 percent R.H.) were apparent from the condition of the specimens at the end of the 6 hour test. In dry air there was a continuous film on both the slider and the disk. At 30-60 percent humidity, metal contacts were seen protruding through the film on the slider. Thus it seems that the increase in friction is due to the increased metal contact, α, resulting from disruption of the lubricant film by moisture. At the highest humidities there was no MoS_2 on the slider; on the disk, however, the MoS_2 had spalled out from between the asperities and had accumulated in front of the slider. The exact reason for the downward trend at high humidities is not known. One possible explanation is as follows: in the short duration of the runs the accumulation of lubricant on the leading edge of the slider may provide some lubrication on the otherwise clean metallic contacts and explain the trend to lower friction values at the highest humidities. There are other factors such as corrosion that could be equally important. Corrosive staining of the steel specimens occurred at the higher humidities.

The rate of wear increased linearly with the availability of more moisture throughout the entire range of humidities. Amount of wear is related to the increased metallic contact that occurred with

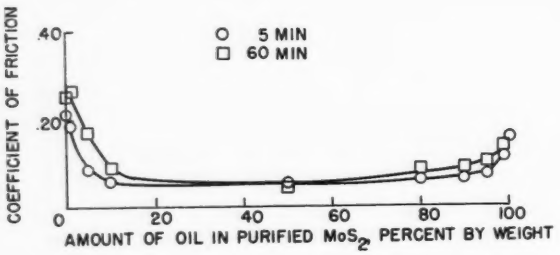

Fig. 6. Effect of oil and MoS_2 mixtures on coefficient of friction for various periods of time. Load, 40 pounds; sliding velocity, 5.7 ft./min.; room atmosphere (40-50 percent relative humidity).

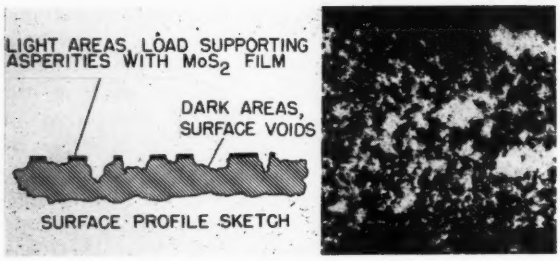

Fig. 7. Photomicrographs of surface of disk after 5 minutes sliding using MoS_2 plus oil as lubricants. X50.

greater humidity.

Effect of Humidity on Other Solids. Tests were then run to check this effect of humidity with other solids and bonded films. The various solids were tested under three different environments: (1) dry air (<6 percent R.H.), (2) moist air (85 percent R.H.) after dry air, and (3) dry air after moist air. These data are shown in Table I. The effect of the moist air after running for ½ hour in dry air was to increase the coefficient of friction for the MoS_2 powder, bonded, and CdI_2 films. When dry air was readmitted the values of the coefficient of friction returned to their original value. These trends were also apparent with PbI_2 but to a lesser degree. No effect was detected with zinc stearate as a lubricant. With graphite this trend was reversed. The high coefficient of friction recorded here for graphite was the result of the initial failure. When graphite was run in moist air first the value of the coefficient of friction was 0.06-0.1.

In view of these experiments with humidity it would be expected that an increase in temperature would drive the moisture out of the film and restore adequate lubrication. This was the case for MoS_2, as shown in Fig. 4 where f is plotted against temperature for two different runs, one at room atmosphere and one in dry air. These data show that as the temperature is increased the friction is lowered to a value approaching that obtained in dry air. Thus, the adverse effect of humidity may be minimized by running at higher temperatures.

Effect of Abrasive Particles in MoS_2. In order to check the effect of hard particles in a softer lubricant various amounts of 200 mesh silica were added to MoS_2 and tests run for 6 hours in dry air. These data are shown in Fig. 5a where the wear area is plotted against silica content. When no silica is present the wear area is very small. With an addition of ½ percent silica the wear increased appreciably. The coefficient of friction also increased slightly with an increase in silica content (Fig. 5b).

Effect of Oil Additions on Lubrication with MoS_2. Various percentages of oil and MoS_2 were run for one hour in order to check that method of application and also determine the role of the oil. Fig. 6 is a cross plot of coefficient of friction versus percent of light mineral oil mixed with MoS_2 showing the 5 and 60 minute values. With small additions of oil, the friction was lowered initially and then increased. With large additions of oil the coefficient of friction approached that for the oil alone.

A photomicrograph of the surface (Fig. 7) after several minutes of sliding shows that the shearing has been confined to the tips of the asperities. Thus, the coefficient of friction obtained with oil-MoS_2 mixtures should be compared with that for the dusted film rather than the continuous film. In comparison of such data (Fig. 2) it can be seen that the coefficient of friction for the dusted film was always less than that for the mixtures of MoS_2 and oil. Thus in applications where minimum friction is desired it would seem that the MoS_2 alone is preferred to the oil MoS_2 mixtures.

Friction of Possible Solid Lubricants. The friction of possible solid lubricants with varied crystal structure was determined in order to check lubricating effectiveness and to further understand the basis of their lubricating action. Results of this investigation are reported in detail in Ref. 5. These tests were run in essentially the same manner as previously described except that three hemispherical riders were substituted for the cylindrical nodes. Friction time curves were obtained for each solid. The one minute and thirty minute values of friction coefficient and wear area are shown in Table II.

Table I. Effect of moisture in atmosphere on coefficient of friction for solid lubricants. Key: (a) When graphite was run first in moist air f was 0.06 to 0.10.

Lubricant	Coefficient of friction, f		
	Dry air	Moist air after dry air	Dry air after moist air
Molybdenum disulphide powder	0.06	0.20	0.06
Molybdenum disulphide bonded film on disk	.09	.22	.09
Molybdenum disulphide bonded film on slider and disk	.26	.34	.31
Lead iodide powder	.27	.29	.27
Cadmium iodide powder	.08	.19	.07
Graphite powder	[a]Fails	.16	.19
Zinc stearate	0.06-0.09	0.06-0.09	0.06-0.09

The criterion selected for comparing solid lubricants was that the best solid lubricant was the one which had the lowest coefficient of friction and the lowest slope on the friction time curve while giving low wear. Using this criterion the best solids tested were MoS_2, zinc stearate, graphite, $CdCl_2$, PbI_2, $CoCl_2$, and WS_2; other layer lattice solids which were effective are HgI_2 and Ag_2So_4. Not all layer-lattice solids were effective lubricants for example $Ba(OH)_2$, TiS_2, and Na_2So_4. Effectiveness as a solid lubricant seemed to be associated with the formation of adherent films on both mating specimens. Some low shear strength solids that did not have a layer lattice were tested and gave initial surface protection. The solids are listed in Table II in order of decreasing effectiveness. The wide variation in friction coefficients obtainable with MoS_2 can be noted by comparing data for double bonded films in moist air (Table I, $f = 0.34$) with that obtained in dry air after 1 minute of sliding (Table II, $f = 0.017$). This result should be characteristic of many solid lubricants.

CONCLUSION. The lubricating effectiveness of solid lubricants varied greatly as the result of various methods of use and conditions of environment. For instance, the friction coefficient for MoS_2 varied by a factor of 20 depending on humidity and other conditions of operation as well as how it was applied. Friction coefficients must be used with much caution when considering solid lubricants.

Lubrication effectiveness can be attributed more to the formation of adherent films on the lubricated surfaces rather than to any particular crystal structure. Humidity, the presence of oil or water and other factors influence the formation of adherent films. Mixtures of MoS_2 and oil must contain at least 10 percent of either to obtain the minimum values of friction coefficient. Minor amounts (0.5 weight percent) of abrasive contaminant increase wear appreciably.

Several materials with layer lattice structure provided effective lubrication: these materials in-

Table II. Solids used in friction and wear investigation. Key: (a) All materials except graphite were run in dried air; (b) Ref. 11; (c) Failed to lubricate.

Material	Purity	Affinity for water (b)	Coefficient of friction		Wear area, sq in.	Observed effect of solid on specimens
			(1 min sliding)	(30 min sliding)		
Effective lubricants						
MoS_2	Possibly slight oxide present	Insoluble	0.017	0.047	0.0016	None
CdI_2	Chemically pure	Soluble	.04	.06	.0023	None
$CdCl_2$	Chemically pure	Soluble	.03	.07	.0019	None
WS_2	Unknown	Insoluble	.05	.08	.0018	None
Ag_2SO_4	Chemically pure	Slightly soluble	.14	.14	Negligible	None
PbI_2	Chemically pure	Slightly soluble	.28	.28	.0018	None
[a]Graphite	Purified commercial lubricant small particle size	Absorbs water	.06	.11	.0034	None
$Zn(C_{18}H_{35}O_2)_2$	Chemically pure	Insoluble	.07	.11	.0032	None
$CoCl_2$	Prepared by dehydration	Soluble	.04	.10	.0020	Slight rusting
HgI_2	Chemically pure	Very slightly soluble	.18	.18	.0021	Appreciable corrosion
$CuBr_2$	----------	Deliquescent	.04	.06	.0021	Considerable rusting
AgI	Chemically pure	Insoluble	.19	.25	.0033	None
SAE 60	----------	--------	.13	.108	.0020	----------
Ineffective lubricants						
$NiCl_2$	Prepared by dehydration	Deliquescent	0.03	0.10-.15	0.0024	Rusting
$Ca(OH)_2$	Chemically pure	Slightly soluble	.18	.2 -.25	.0026	None
$Mo(OH)_2$	Chemically pure	Soluble	.32	(c)	-----	None
TiS_2	Unknown	Hydrolysis	.20	(c)	-----	----------
I_2	----------	----------	.15	.30	-----	----------
$HgCl_2$	Chemically pure	Soluble	.32	.38	-----	Appreciable corrosion
$PbCl_2$	Chemically pure	Slightly soluble	.31	.45	.0029	None
AgCN	Unknown	Insoluble	.02	- -	.0036	----------
CuCl	Chemically pure	Slightly soluble	.38	.37	.0039	Slight rusting
Na_2SO_4	Chemically pure	Soluble	.36	(c)	-----	None
Fe_3O_4	Chemically pure	Insoluble	(c)	- -	-----	None
BN	Technical grade	Insoluble	(c)	- -	-----	None
NiO	Chemically pure	Insoluble	(c)	- -	-----	None
Mica	Technical grade	Insoluble	(c)	- -	-----	None
Talc	Unknown	Insoluble	(c)	- -	-----	None

clude WS_2, CdI_2, $CdCl_2$, $CoCl_2$, Ag_2SO_4, PbI_2, and HgI_2. They were not as effective as MoS_2 but were at least as effective as zinc stearate and graphite in these tests. Layer lattice crystal structure could not be used as the only criterion for selection of solid lubricants.

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COMMENTARY by A. Dobry (Westinghouse Research Laboratories). The authors have presented some very intriguing experimental results. We were particularly interested by their finding that boron nitride is not a lubricant, in spite of its layer lattice structure. We, as well as a number of other investigators, have obtained this same result. However, we have recently heard that a new, ultra-pure form of boron nitride is now available and that this pure material does act as a dry film lubricant. If true, this would imply that the lubricity of boron nitride is markedly dependent on its purity.

When the surfaces are lightly dusted with MoS_2, the friction is less than with a thicker film. The authors suggest that this is due to the amount of lubricant being insufficient to fill the interstices, confining the sliding to the tips of the asperities. But, if this were the case, one would expect that the MoS_2 would soon be rubbed off the high points and would collect in the valleys. The rubbing surfaces would then have metal to metal contact and the coefficient of friction would rise to that of dry friction.

It seems simpler to attribute this decrease in friction to a thinning out of a continuous MoS_2 film. This results in the flow pressure of the film approaching that of the asperities, and, as the authors have pointed out, the coefficient of friction then declines. The difference in hardness between the asperities and the bulk material would seem to be irrelevant here. Since the flow pressure of the film will only begin to approach that of the underlying stratum at points where the film thickness is small, that is, at the asperities, the strength of the bulk material will be immaterial as long as it is greater than the flow pressure of MoS_2.

COMMENTARY by F. T. Barwell (Mechanical Engineering Research Organization). The authors have produced an interesting set of results from experiments on various solid lubricants, and their demonstration of the contrasting effects of moisture on the friction of molybdenum disulphide and of graphite are of particular value.

The authors mention that crystal structure is of less importance than the ability to form adherent films on the surfaces. It is well known that the effectiveness of many fluid lubricants is dependent on their ability to react chemically with the surfaces. Some recent experiments on lead sulphide at Mechanical Engineering Research Laboratory (Glasgow) seem to indicate that even with solid lubricants such reactions may be initiated by the local pressure and/or temperatures generated at the contact and may under certain circumstances play an important role in the lubrication. Have the authors any evidence of whether any such reactions were taking place under the conditions of their experiments, or have they otherwise elucidated the nature of adherence in these films?

It would have been interesting to learn a little more about the load capacity and endurance of the solid lubricant films studied, for these are probably the factors of greatest practical importance. Some earlier work of our own (Scientific Lubrication, Sept. 1951, Vol. 3, No. 9, p. 10) did show that the conjunction of a solid lubricant with a prior surface treatment, such as MoS_2 bonded to phosphated steel, greatly enhances the characteristics of the lubricant film; while more recent work has shown that "sulfinuzation" of the steel surface before the application of MoS_2 may be equally effective.

The beneficial effects of this double treatment are readily understandable in the light of general ideas on friction such as those discussed in the paper. The function of carrying the load and reducing the friction is performed by the solid MoS_2 film, but in those regions where the film is momentarily penetrated the surface treatment may still be able to prevent welding of the opposing surfaces. If a complete film can be maintained, such a combination is reasonably effective even with a notoriously difficult material such as titanium. The accompanying table shows the magnitude of the improvement in load capacity obtainable, under conditions similar to those in previous tests (reference given above); the endurance of these films has not yet been studied.

Materials	Surface Treatment	Lubricant	Seizure load - lb.
Commercially pure titanium against Commercially pure titanium	None	Mineral Oil	5
	None	Bonded MoS_2	80
	None	Bonded MoS_2 + mineral oil	60
	Phosphated	Mineral Oil	120
Commercially pure titanium against Titanium alloy 150A.	Phosphated	Mineral Oil	35
	Phosphated	Bonded MoS_2	165
	Phosphated	Bonded MoS_2 + mineral oil	105

The factors leading to the formation of low-friction lubricant films with solid lubricants now seem to be well established. It is to be hoped that the authors will continue their interesting investigation to include also the factors leading to films of greater endurance.

COMMENTARY by I-Ming Feng (Massachusetts Institute of Technology). The authors are to be commended for their interesting and valuable contribution to the literature on solid lubricants. The systematic fashion in which they made their study concerning the effect of moisture on coefficient of friction of solid lubricants could be extended to the investigation of the effect of various condensable vapors.

The authors mentioned that a thick bonded MoS_2 film

on metal base gives a higher coefficient of friction than a comparatively thinner bonded film because the flow pressure p is influenced by the thickness of the film. The effect of flow pressure p on the friction coefficient of MoS_2 film has also been studied by the writer. (Kennecott Research Progress Report No. 1 & 2; M.I.T., 1950.) The measurement of the friction of bulk MoS_2 was made first to establish a basic value of reference with which other measurements of friction of various MoS_2 coatings can be compared. The compressed MoS_2 pellets were prepared by the Laboratory of the Climax Molybdenum Co. The friction coefficient of bulk compressed MoS_2 against bulk compressed MoS_2 was found to be 0.20. The friction coefficients of bulk compressed MoS_2 against various materials of different hardness (harder than the MoS_2 pellet) were also determined. They are in the range of 0.20 to 0.22. This demonstrates that frictional coefficient depends only on the flow pressure of the softer member of the rubbing pair. With a ¼" diameter steel ball rubbing against a thick MoS_2 film prepared on a SAE 1020 steel disc with a binding material, the friction coefficient was found to be essentially the same as that of compressed MoS_2. Experiments were then carried out with ¼" steel balls sliding against thin rubbed-on MoS_2 films on metal bases of various hardness. The results are tabulated in the accompanying table.

Base Metal	Hardness Rockwell "F"	Kinetic Friction Coefficient	
		Initial Valve	Leveling-off Valve
Brass.	28	0.055	0.105
SAE 1020 Steel	67	0.050	0.095
Cu-Be Alloy	80	0.045	0.07

AUTHORS' CLOSURE. We are grateful to the discussors for their comments and interesting supplemental information. Mr. Dobry has interjected a searching comment regarding "ultra-pure" boron nitride. We are not familiar with this product and hence have no data on its performance.

Surface studies have indicated that the condition shown in the photo-micrograph of Fig. 7 occurs with other methods of applying the MoS_2 when a small amount of the solid is used. It (Fig. 7) is an accurate representation of the condition observed when MoS_2 was employed as a dusted film. In the case of the dusted films the total contact area was very small and the film appeared continuous in the localized areas of contact. The film thickness was so small that it was unlikely the flow pressure of the film material could have influenced the shear area. With the extremely thin films the hardness of the asperities would then determine the contact area. Data of Fig. 2 shows that these very thin films were not easily ruptured or worn away. Greater metallic contact would have been indicated by increasing friction during the run.

NACA experience is consistent with the suggestion of Dr. Barwell that chemical affinity of solid lubricants is important in forming adherent films. Electron diffraction studies were not successful in identifying any chemical reaction products that might occur in lubrication of steel with MoS_2. Many of our results, however, strongly indicate that chemical effects of the solid lubricant on the lubricated metal or its oxides contribute to the formation of adherent films.

Pretreatment of metal surfaces by methods such as phosphating is known to increase endurance of solid lubricant films. Surface roughness of the base metal is also important to the endurance of solid lubricant films. The phosphate treatment serves a dual role. It provides surface characteristics (roughness) conducive to good mechanical bonding with resins, and also the phosphate treatment produces a reaction film that by itself has solid lubricant properties.

The influence of hardness of the base metal on friction with thin MoS_2 films was shown very effectively by Dr. I. Ming Feng. Dr. Feng's data and similar NACA data on steel surfaces are in good agreement.

New Products

Gits "HH Type" Shaft Seal. Effective shaft sealing in absolute minimum space (both radial and axial) under extreme operating conditions of temperature, pressure, and seal face surface speed, are featured in the new "HH Type" shaft seal. Principal factors creating such flexibility are a variable pressure balance feature and a new, patented method of incorporating synthetic rubber, silicone, teflon, and Kel-F. (Gits Bros. Mfg. Co., LE-11/5, 1866 S. Kilbourn Ave., Chicago 23, Ill.)

Unitized ASTM-Type Vacuum Column. For laboratory convenience the vacuum-distillation column introduced last year has been redesigned into a self-contained unit mounted in a metal shield; facilities are provided for rapid access to all parts requiring removal or adjustment during operation. Originally developed to coordinate with a standardization program relating to the use of ASTM-Type distillation for the characterization of lubricating oils and reduced crudes, it is otherwise the same as that produced for Myers & Kiguchi of C. F. Braun & Co. The apparatus includes two condenser jackets, one surrounding the delivery tube leading from the column to the product receiver, and a second vertical knock-back condenser for lighter materials; the distilling flask is of 500-ml capacity Vycor-brand glass and equipped with a thermometer well and 35/25 socket joint. The silvered vacuum jacket is protected against thermal shock and expansion by internal-type expansion bellows, while the internal silvering on the vertical condenser is provided with a ¼-inch wide full-length window; a 10/30 standard taper joint is included at the top of the condenser for the thermocouple. (Glass Engrg. Labs., LE-11/5, 564 O'Neill Ave., Belmont, Calif.)

"Glas-Kleer" Filter Cartridges. Designed to meet the need for a low-cost, scientifically graded throw-away filter cartridge, "Glas-Kleer" cylinders have been developed from super-fine glass fibers chosen for their high chemical and temperature resistance and non-absorbent structure, and are being offered in 2, 10, 25, and 50 micron grades. Each cylinder is lined with fiber glass cloth to insure lint-free filtration and is provided with variable density to insure long service life and high holding capacity. (Porous Plastic Filter Co., LE-11/5, 30 Sea Cliff Ave., Glen Cove, N. Y.)

(Continued on p. 353)

Factors Influencing Humidity Cabinet Test Results

by H. A. Hartung*

It is found that the specification of 95-100% Relative Humidity is not sufficient to insure constant rusting conditions in a humidity cabinet. Much better precision is obtained when a cloud is used as the rusting environment. A simple device for producing such a cloud is described, and a number of test results are discussed. The influence of panel preparation and type on precision is shown to be considerable, but ordinary sand-blasted panels are capable of giving satisfactory results.

Most early humidity cabinets were disarmingly simple devices. Perhaps this simplicity led experimenters to hope that cabinet tests could be designed in a simple way. Experience has shown that this is not usually possible. Even with close control over a large number of variables affecting the test, the scatter in results is disconcerting.[1] It is not possible for one laboratory to check with another except in a most general way. Worse than this, it is frequently impossible for one laboratory to check its own results quantitatively in a given cabinet. Worst of all, in a single test where multiple specimens are involved the spread in life among members of a set is often uncomfortably large. To illustrate the problem, Fig. 1 shows some typical results obtained with conventional practice and equipment in a single cabinet over a period of a year or two. The data plotted include variations within a given set of panels (precision) and also variations observed in repeat tests on the same rust-preventive materials (repeatability). Both precision and repeatability are poor, deviations from the average of 40% being common.

Recently a program has been proposed within ASTM for attempting to improve the technique of humidity cabinet testing, with increased control over the preparation of test specimens as an initial step. While proper control over all variables is essential, it appeared that there must be an underlying fault in the mechanics of humidity cabinet testing to cause such consistent deviations as occur. That others hold this to be true is evidenced by the large number of substitute tests that have been developed for the conventional cabinet.[3,4,5,6] A program was set up in the author's laboratory to study the test and to attempt to improve its precision and repeatability. Three aspects were considered: The cabinet proper, test panels and their preparation, and the conditions under which the test is conducted. The present paper discusses some of the results found in the first two of these groups; the study is still in progress.

HUMIDITY CABINET. Test results obtained in the past in a cabinet without a rotating stage gave several important clues as to what might be a worthwhile approach to the problem. First, under many conditions it was noted that panels close to the walls of the cabinet were the first to fail in a given set. Again, average test life for a given product varied considerably with the time of year and with weather conditions. Finally, when this cabinet was placed in a constant temperature air-conditioned room, it was markedly less severe than when it was in ordinary surroundings. From these observations it was apparent that conditions outside the cabinet affected conditions inside it. Relative humidity measurements on air leaving the cabinet showed it to be at least 95% R.H. at all times. Apparently, then, the specification of 95-100% R.H. is not sufficient to insure uniform rusting conditions within the cabinet.

In such a cabinet the walls are usually cooler than the interior of the cabinet. Changes in ambient temperature affect the temperature of the walls of the cabinet, inducing some temperature cycling inside the cabinet. Variations in the frequency and degree of this cycling affect rusting conditions within the cabinet. Furthermore, panels ad-

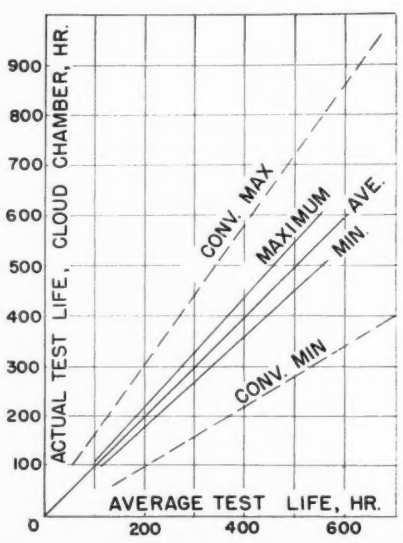

Fig. 1. Spread in test life as a function of average test life in a conventional humidity cabinet (dotted) and experimental cloud chamber (solid).

*The Atlantic Refining Co., 260 S. Broad St., Philadelphia, Pa.

(This paper was sponsored by the ASLE Technical Committee on Physical Properties of Lubricants, and presented at the ASLE 10th Annual Meeting, Chicago, April 14, 1955.)

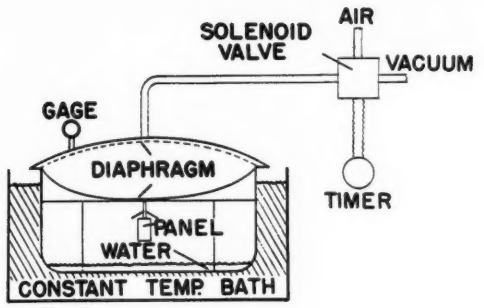

Fig. 2. Construction of experimental cloud chamber humidity cabinet.

jacent to the walls are most affected in temperature by these cycles, and they show the greatest tendency to rust.

At this stage a small cycling temperature cabinet was built to study this matter more closely. The concept of cycling temperature in humidity cabinet testing is not new. Schlossberg,[2] for instance, in his summary of the problem lists a number of devices of the cycling temperature variety. In the cabinet built for this program, it was desired to have the air temperature cycle in a controlled manner. Pressure variation of the cabinet atmosphere was used to achieve this temperature cycle easily, rapidly, and reproducibly. Fig. 2 is a sketch of the equipment. Note that the thin rubber diaphragm separates the test chamber from the actuating chamber so that the air in contact with the panels is not continuously being changed, and no separate saturator is needed. In operation the unit is assembled with the diaphragm in some intermediate position. Pressure applied to the actuating chamber by air expands the diaphragm, raising the pressure and temperature in the test chamber. The actuating chamber is then connected to a vacuum line, causing the diaphragm to move up to the top of the vessel and rapidly lowering pressure and temperature in the test chamber. If the test chamber air is saturated at the higher pressure (by water in the bottom of the vessel) and if the expansion is great enough (about 30-35%), a cloud is formed much as in the celebrated Wilson Cloud Chamber. This is the condition desired: To substitute for the usual air of 95-100% R.H. an atmosphere that is saturated and

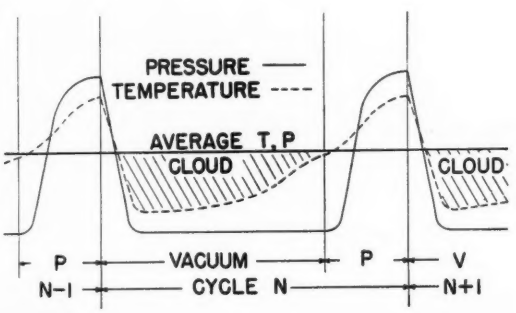

Fig. 4. Cyclic temperature and pressure variations in experimental cabinet.

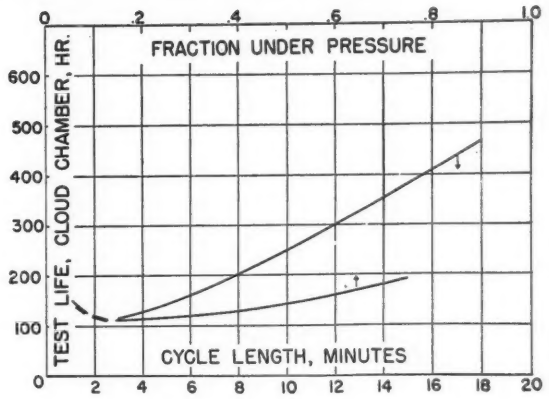

Fig. 3. Influence of cycle length and split on test life in experimental humidity cabinet.

contains minute water droplets in suspension.

Operating under conditions that would form a cloud, +4 to —3 psig, a number of interesting observations were made. The rusting of blank panels and those coated with mild rust preventives was extremely uniform. With more effective rust preventives tests were terminated after initial failure occurred, as the time required for complete rusting would be too great. In all cases precision was much improved over the conventional cabinet (Fig. 1). Shorter cycles were found to be more effective than longer cycles in promoting rusting (Fig. 3), and precision was poorer when the cycles were long or very short. As cycles become very short, test life increases. The exact shape of the curve depends on the rate at which the system responds to the pressure changes applied to it. With short (3 minute) cycles, it was found that severity increased as the fraction of the cycle spent under pressure decreased (Fig. 3), again to a fraction dependent on the mechanical arrangement.

These results lend weight to the theory that a cloud is a more effective and uniform rusting medium than air of 95-100% R.H. Measurements taken in the test chamber show that the air temperature falls as the pressure falls, but that it soon rises as

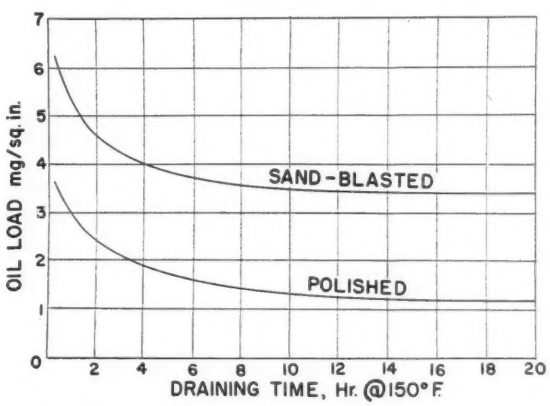

Fig. 5. Draining curves at 150 F. for oil of 100 SSU @ 100 F.

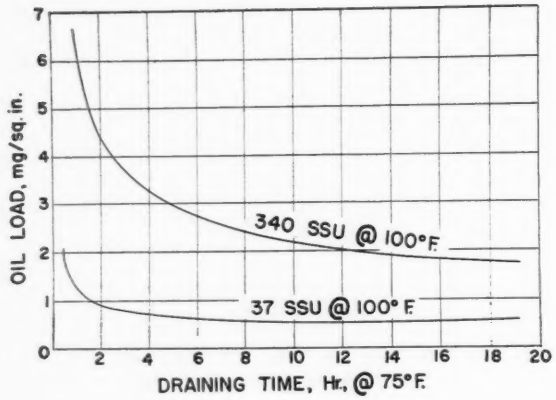

Fig. 6. Drainage of two oils at 75 F. from polished steel panels.

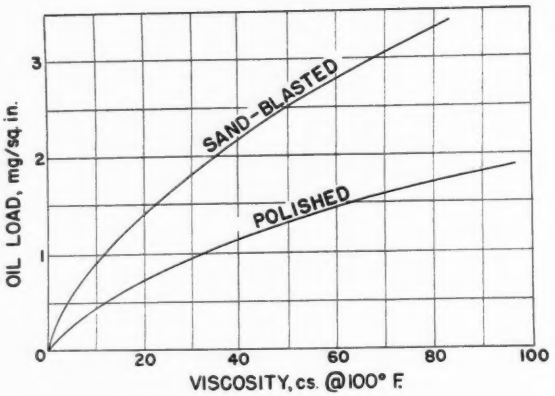

Fig. 7. Equilibrium surface oil load at 75 F. as affected by oil viscosity (drained 28 hr. at 75 F.).

the metal parts lose heat to it. Fig. 4 shows this behavior; portions of the cycle where a cloud exists are shown shaded. It can be seen that longer cycles and a higher proportion of the cycle under pressure would reduce the net time during which the cloud is present in the test chamber.

Baker, Jones and Zisman[6] have described a "Fog Cabinet" in which a dispersion of water is introduced as a fine spray into the test chamber. This is somewhat similar in effect to the cabinet described here, although their apparatus is basically different and considerably more complex. The dispersion formed in their "Fog Cabinet" by atomization is probably coarser than that produced by the mechanism described here. Baker et al. found better precision for their device than for a conventional humidity cabinet. They also report good correlation with a humidity cabinet on the basis of weight loss of uncoated panels. This is a rather different matter than penetration by water vapor of a protective coating; there is some evidence that penetration of certain coatings is a diffusion process, rather than one of leaching out the polar additives as Baker and co-workers propose.

A number of other studies are being made in this experimental cabinet which has proven to be a very useful device. It is of simple construction, rugged, cheap, and it gives very uniform results. It shows some acceleration over conventional cabinets, but seldom by more than a factor of two. Despite the fact that the desire here was to study basic phenomena rather than to develop an accelerated test, the cabinet described seems to have considerable merit as a laboratory evaluation tool.

TEST PANELS. Virtually all the tests described in this paper were made with sand-blasted low carbon steel panels 1" x 2" x ⅛". While these are far from being ideal as test specimens, they represent a good point of departure for judging the effect of changes in panels and their preparation. Any change that narrows the spread between the maximum and minimum lines of Fig. 1 can be considered an improvement.

Coating the panels for testing offers many opportunities to worsen the precision and repeatability of the test. The effects of draining time and surface roughness on retention of oil-type rust preventives are shown in Fig. 5. It is apparent that the grit size used in producing a sand-blasted finish will influence the oil load. Similarly, if panels are drained at elevated temperature, then tested at some lower temperature, draining time can affect the amount of oil left on the surface. In Fig. 6 drain-

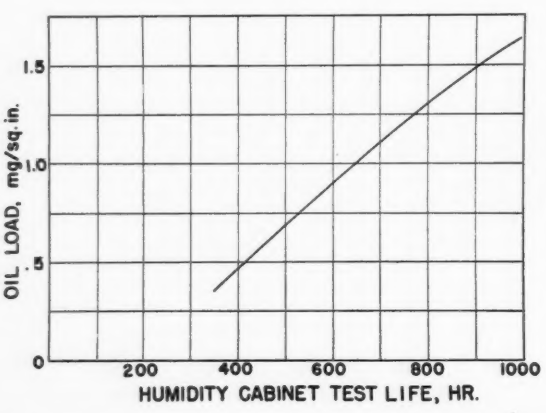

Fig. 8. Relation of humidity cabinet test life to oil load.

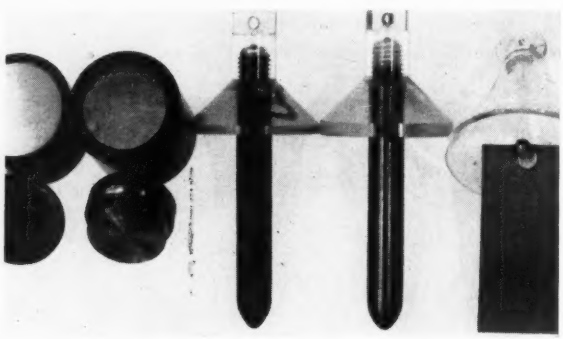

Fig. 9. Specimens tested in experimental humidity cabinet: (left to right) steel buttons of two different thicknesses molded in bakelite, sand-blasted and polished ASTM Turbine Oil Rust Test specimens, sand-blasted rectangular panels.

ing curves at room temperature for two oils of different viscosity are shown. Here it is obvious that if different additives or concentrates are to be compared, the comparison should be made at the same viscosity to give the same oil load. Fig. 7 shows how the equilibrium oil load (drained 28 hours at 75 F.) varies with viscosity for two different finishes. The importance of viscosity and oil load is brought out in Fig. 8. Oil blends which varied only in the viscosity of the base stock were tested under identical conditions, giving improved protection as oil retention increased.

These results are elementary, but it must be remembered that rather minor differences in surface finish, oil viscosity or draining temperature and time can affect the cabinet life of oil-type rust preventives. Part of the spread shown in Fig. 1 is believed to be due to incomplete control over these variables.

Test specimens of a number of different shapes and sizes have been evaluated in the experimental cabinet. Some of these are shown in Fig. 9. The bakelite molded buttons were tried with the exposed surfaces vertical and horizontal, facing up. These buttons were polished, and blanks of two different thicknesses were used. The ASTM Turbine Oil Rust Test specimens were used polished and sand-blasted. Sand-blasted rectangular panels were used for the bulk of the work, although a few tests on polished plates were run. Summarizing the work on these and other specimens, it can be said that polished panels gave poorer precision than sand-blasted panels. There was no particular effect observable due to differences in specimen size, although the only direct comparison was made with the polished buttons. Although no panel size, shape, or finish gave better precision than sand-blasted rectangular panels, work is continuing, and there is presently under study a method which shows promise of being an improvement. In this technique the test surface is generated in the presence of the rust preventive in such a way that contamination from the atmosphere is eliminated. Although the improvement is marked, it is not as important to good testing as that resulting from the improved rusting atmosphere.

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ACKNOWLEDGEMENT. Most of the experimental work described in this paper was done by W. E. F. Lewis and J. Abramchuk. The author is especially indebted to J. W. Johnson and A. C. Smith, Jr., for discussion and helpful criticism of this work.

COMMENTARY by D. Bootzin (Rock Island Arsenal). The paper presented by Mr. Hartung is particularly interesting to Rock Island Arsenal since humidity cabinet testing is employed quite regularly for evaluating corrosion preventive compounds submitted for qualification under government specifications, and as an important test used in research and development work.

It is agreed that uniform or constant test conditions are desired and that the humidity cabinet test currently encompasses a host of variables which need to be analyzed as to their effect on the accuracy of results so that they can be either eliminated or tolerated.

Mr. Hartung discusses a number of the variables including how conditions outside of the cabinet affect panel location inside the cabinet, the consideration of the physical nature of the moisture involved, whether spray or vapor; the test panel preparation as related to oil retention which is dependent on draining time and surface roughness which in turn is dependent on temperature, viscosity, grit size of sand and abrasive paper employed.

All of these to some extent affect the repeatability where the same operator repeats the test and reproducibility where different operators or different laboratories perform the test and thereby account to a large degree for the disagreement between laboratories as to results.

Many of these variables obviously can be eliminated in comparative tests, once they are recognized, e.g. environmental influence on the cabinet can be minimized by a constant temperature room, panel location in a cabinet be deemphasized by use of a rotating stage, etc.

Mr. Hartung's use of a cyclic cloud-producing device apparently provides a test condition which has considerable merit; however, it appears that it produces a specific fog or cloudy condition and in that respect is no better than another type of humidity cabinet which produces a constant vapor or water spray.

The fact that no air change is involved appears to be unsatisfactory since any volatile inhibitor would provide protection to other panels which would be tested in the same atmosphere.

The test provides a leaching action since considerable condensation occurs which may not be desirable in all cases, and the amount of oxygen in an enclosed space could be depleted to the extent that would reduce normal corrosive action.

The test also does not simulate any actual environmental test condition in that a periodic vacuum is introduced which may produce unrelated physical and chemical changes on the protective films.

The variable which was not mentioned, and which causes considerable difficulty, is interpretation of what constitutes failure or initial corrosion. The number and size of rust dots as detected by different operators varies greatly and is reflected in tabulated data.

Finally, even though the cloud chamber is a fine development, it poses difficulties where a large number of tests are required.

Since these comments are based on the presented paper, which of necessity cannot be too detailed, it is hoped that a complete report will be made available since every effort toward understanding the factors influencing humidity cabinet test results is greatly appreciated.

COMMENTARY by H. L. Leland (Esso Research & Engineering Co.). Anyone who has had the occasion to conduct humidity cabinet tests on rust preventives will endorse

the author's statement that "there must be an underlying fault in the mechanics of humidity cabinet testing." The repeatability and reproducibility of humidity cabinet tests are undesirably large, in spite of the precautions taken in most laboratories to standardize their operating techniques. The cabinet proposed in Mr. Hartung's paper appears to minimize some of the errors inherent in the familiar JAN-H-792 cabinet and shows promise as a more precise test method.

The effort to develop improved test methods for rust preventives now under way in Section B of Research Division X of ASTM Committee D-2 has yielded knowledge that supports Mr. Hartung's belief that improvement in the test enclosure and environment is necessary. Designed round-robin experiments have shown that variables in sandblasting, panel cleaning, panel drainage time below cabinet temperature, and panel storage were unimportant compared to some factor or factors not yet determined. It is highly likely that these factors are connected with the operation of the JAN-H-792 cabinet. The round-robin tests also showed that sandblasted panels give more reproducible results than polished panels and that the two test oils selected to give a short life and a long life, respectively, were clearly differentiated. The ASTM program is now being aimed at cabinet operation.

Mr. Hartung has shown the influence of oil viscosity, panel finish, draining time, and draining temperature on oil retention on the test specimens. This in turn is related to humidity cabinet test life. It would be interesting to know whether the same general relationships apply in the cloud cabinet. Many rust preventives protect in the JAN-H-792 cabinet largely by forming durable emulsions. The lack of wide cycles in the temperature and humidity conditions in this cabinet does not cause emulsion collapse. This condition is highly unrealistic in terms of field performance. The Cloud Cabinet, with temperature and pressure cycles, may cause emulsion collapse and thus result in the shorter life periods noted in the paper.

One of the most interesting features noted was the protective caps over the specimens shown in Fig. 9. In our laboratory we have noted that improvements in both humidity cabinet life and precision were obtained when a plastic cover was placed on the top of the test panels to prevent droplets from falling directly on this surface. It is possible that some of the improved precision noted in this paper was a result of using these protective caps. Similar gains in precision can be obtained by enclosing test specimen in vented bottles in the JAN-H-792 cabinet.

The author is to be commended for an interesting approach to this complex problem. The deficiencies of the present procedures are well known. Perhaps the complete revision of the test chamber as proposed here will provide a more satisfactory tool for laboratory evaluation.

AUTHOR'S CLOSURE. Dr. Bootzin raises a number of points about the operation of the cloud chamber which some further discussion may help to clarify.

Experiments indicate that the life of a panel in the cloud chamber is not affected by the "intensity" of the cloud, but only by the length of time the cloud is present. If the cycle is adjusted to give a cloud for the maximum time, increasing the expansion by raising pressure and vacuum does not change the severity. Consequently, it appears to be easy to achieve and maintain constant test conditions in the cloud chamber despite changes in operating variables.

With regard to oxygen depletion, none has been detected (by loss of pressure) in operating periods as long as 72 hours without opening the chamber. Normally the chamber is opened daily to inspect the test specimens. It is true that this method of operation would require isolation of panels protected by volatile inhibitors. On the other hand it offers advantages in the use of special corrosive atmospheres.

Leaching action has not been observed to take place in the cloud chamber under the conditions studied. Condensation is fine and dropwise, and until rust spots appear there is little tendency for the small droplets to aggregate. The effect of the periodic vacuum on the protective films is not known, but can be supposed to be small since the reduction in pressure is only about 2.5 psi below atmospheric. This is a matter of convenience; the cloud chamber could be operated between the limits of 6 or 7 psi gauge and atmospheric pressure to get the required expansion.

The precision of results in the cloud chamber is such that we can substantially reduce the number of multiple specimens that must be used. This fact along with the acceleration of the test makes the capacity of the cloud chamber relatively large. Further, cost of the equipment is low so that for a given investment in money, space, or time, the output of test results from the cloud chamber looks attractive.

The author is in complete agreement with Dr. Bootzin's remarks on the reading of test specimens. We have found the system devised at Rock Island Arsenal to be most satisfactory and use it exclusively for rating specimens.

Dr. Leland points out that the work undertaken by Section B of Research Division X of ASTM Committee D-2 bears out the conclusion that improvement in the test enclosure and environment is needed in humidity cabinet testing. It is hoped that the results presented here may be of some assistance in the future work of this group.

Studies of the effects of oil viscosity, panel finish, draining time and temperature, etc., on cloud chamber test life are not complete. Early indications are that the effects of these variables on test life are similar to those found in the humidity cabinet.

The protective caps shown served the dual purpose of suspending the specimens and protecting them from dripping off the supporting grid. When panels are not hung from this grid, the need for these caps is reduced as there is no path for condensation to run down onto the specimens. Drips can largely be eliminated by placing two or three sheets of filter paper over the grid. Nevertheless, some protection for the tops of panels is still recommended.

The author is grateful for the stimulating comments of the discussors. This discussion has, it is believed, added greatly to the value of this contribution in promoting thought and further research on the mechanics of testing rust preventive compounds.

Book Reviews

The Prevention of Occupational Skin Diseases, by Dr. Louis Schwartz (formerly Medical Director of the U. S. Public Health Service); Association of American Soap & Glycerine Producers, Inc. (Publisher), 295 Madison Ave., New York 17, N. Y.; free individual copies, bulk quantities at cost.

"Diseases of the skin are taking a far larger toll of the nation's industrial output, in terms of lost productive time and compensation payments, than all other occupational diseases combined. The cost to industry of the average industrial worker hit with a skin disease is: 10 weeks off the job, $100 in compensation payments, and $90 for medical care; totaling far more than $100 million annually." Stressing the vital need to safeguard industrial workers from actual or potential skin irri-

(Continued on p. 348)

Maintenance Of Quality In Aluminum Sheet Rolling Oils

by Dr. J. O. McLean*

In the early stages of sheet rolling the dirty oil was treated with concentrated sulfuric acid, centrifuged and re-used. This method proved unsatisfactory. A system was then adopted of using the oil until it became badly contaminated, dumping a portion and adding new oil. This process was wasteful and was abandoned. The next step was to send the dirty oil out to a jobber who purified and returned it. This proved satisfactory but was expensive. Finally an elaborate by-pass filter system was installed based on the use of adsorptive clay. This system, although possibly not the cheapest or best, is in use today. It keeps the oil fairly clean constantly, but occasions excessive loss of additive and requires large quantities of clay which, in turn, causes fairly large oil losses through discarding oil soaked clay.

This paper deals only with those oils used for cold rolling of aluminum sheet which normally lies in the range of 0.100" to 0.006" thick. In general, such oils consist of a major percentage of mineral oil of 40-45 SSU at 100 F. and a minor percentage of an additive. The latter usually is essentially an ester-type compound (for example, fatty material, glycerides, etc.), although other materials have been used with some success.

Maintenance of quality means keeping the oil in such condition that top grade sheet can be produced. The importance attached to this process is indicated in part by:

(1) High cost of equipment to keep the oil in good condition.

(2) One mill may have a 20,000 gallon capacity.

(3) One plant may produce a million pounds of sheet per day, some rolled several times.

In one plant, purifier costs are about $170,000.00. By enabling the oil to be used over and over, however, the purification system has already paid for itself many times. The principal factors controlled are as follows: Specifications of rolling oil components, additive concentration, ash content, viscosity and staining tendencies.

Specifications on Components. Quality starts with the purchase of the rolling oil components, which are bought against rigorous specifications. These include the usual tests on the petroleum component plus distillation and stain. On the additive portion, melting point, acid, saponification, and iodine numbers normally assure us of quality and uniformity.

Additive Content. Additive content is critical. If it becomes too low, rolling characteristics are unsatisfactory; if too high, staining characteristics become objectionable. Normally, standard ASTM saponification numbers are run every other day, and additive is adjusted in accordance with these results.

Ash Content. The ash content is also critical. Materials contributing to ash are principally aluminum and aluminum oxide from the product being rolled and materials such as tramp greases and dirt picked up by the oil from outside sources. The goal is zero ash content; however, a value of not more than 0.005% has been found satisfactory.

Ash content is controlled by an elaborate and rather complex by-pass filtration system using adsorptive clay as the purifying medium. This is the critical component in the quality control system and will be dealt with in greater detail.

Viscosity and Staining. Changes in viscosity are normally associated with external contamination, such as oil from bearings on the mills. If these oils are heavy petroleum products, they contribute to excessive staining of the sheet. If the viscosity becomes high, a distillation is run which shows the presence of heavy oils. If the percentage of bearing oil gets too high, it is necessary to dump the entire batch, since it is ordinarily impractical to remove the oil. This rarely happens. The filter system installed tends to remove stain-producing bodies associated with lubricating oils, hence, after several passes through the clay, staining tendencies are decreased. The best solution to this problem is to prevent stray oils from entering the system.

Greases also cause staining; however, they appear to be removed by adsorption in the filter system.

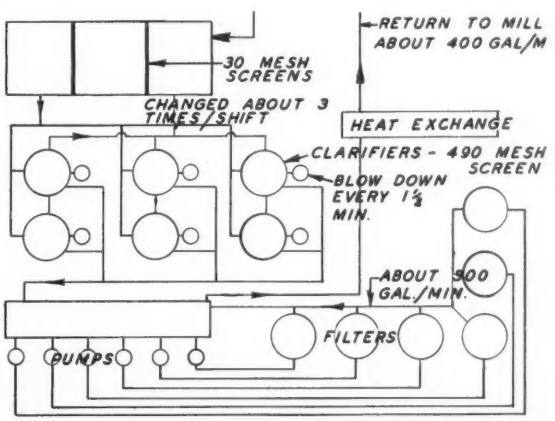

Fig. 1. Schematic diagram of purification system.

*Reynolds Metals Co., 3rd & Grace Sts., Richmond, Va.

(This paper was co-sponsored by the ASLE Technical Committees on Fluids for Metal Working, and Lubricant Reclamation & Disposal, and presented at the ASLE 10th Annual Meeting, Chicago, April 13, 1955.)

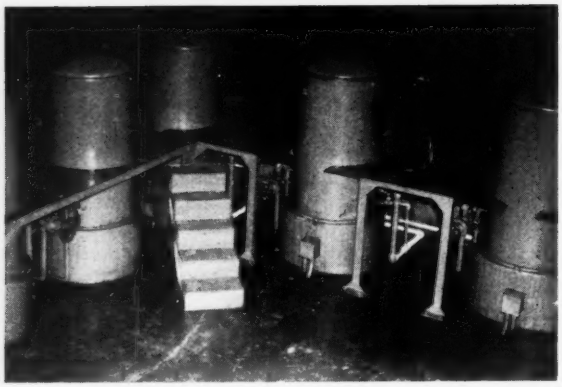

Fig. 2. Oil filters.

There is evidence that iron or aluminum salts form during the rolling operation, and these materials also have staining characteristics. Fortunately, they also are removed by clay filtration.

Description of Filter System. The plants initially reclaimed sheet rolling oils by a complex system involving treatment with concentrated sulfuric acid, water washing, and centrifuging. This method soon proved unsatisfactory, as the acid affected the base stocks and additives, and sometimes entered the clean oil reservoirs.

The next approach was to run with a batch of oil until contamination interfered with rolling. The oil was then dumped. This method was not only expensive, but gave non-uniform sheet.

Another approach was to withdraw the oil periodically from the system and ship it to an outside jobber who reclaimed the material and returned it for reuse. This was satisfactory insofar as performance went, but the cost was excessive.

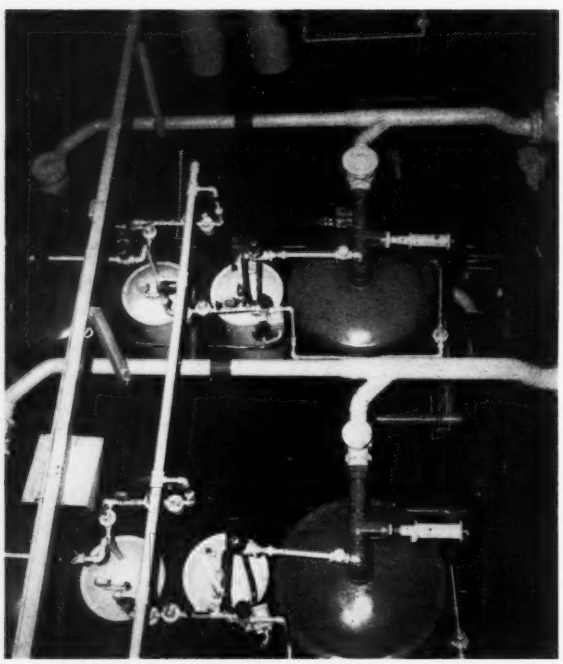

Fig. 4. Looking down at the pre-filters.

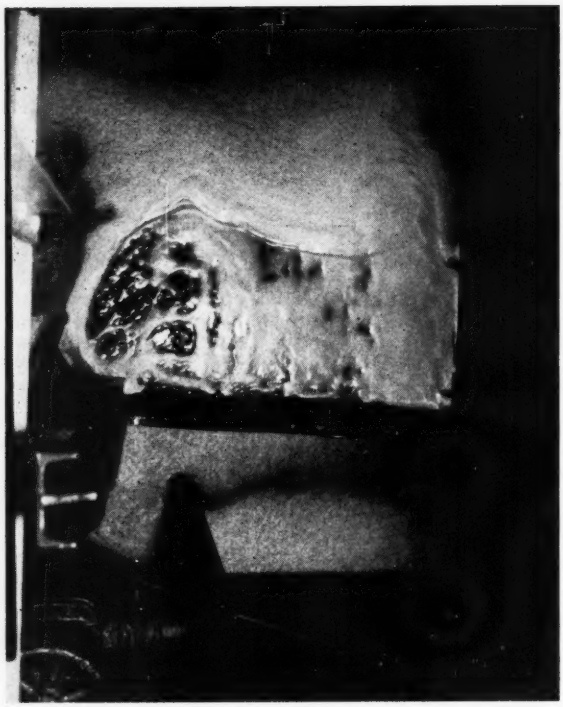

Fig. 3. Looking down into the oil strainers (open concrete pit).

The system next tried was a continuous bypass type of filtration, using active clay as the filter medium, and was based on the principle of purification by adsorption. This technique, still in use today, is expensive to install, requires considerable attention, and is quite wasteful of both mineral oil and additive. At one of our plants a procedure designed to eliminate most of the undesirable features of this method has been installed, and may eventually be used in place of the present operation particularly where new equipment is required.

Fig. 1 shows the present general equipment layout and flow pattern with relative flows indicated. Fig. 2 indicates an actual filter bank used for one mill. Fig. 3 shows the oil strainers which remove the larger solid impurities. Fig. 4 shows the pre-filters (actually fine mesh screen) with automatic blowdown devices, the collected dirt being returned

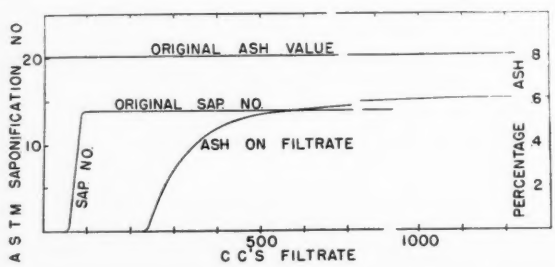

Fig. 5. Clay treatment of used rolling oil.

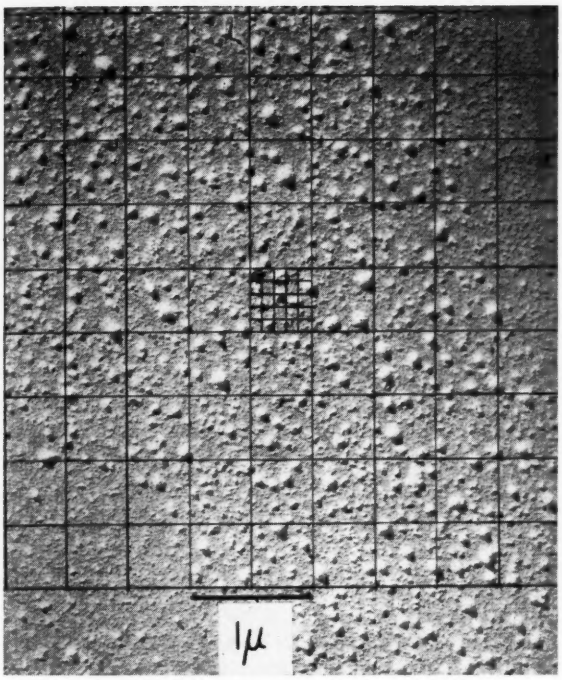

Fig. 6. Electron micrograph of used roll oil (21,700X).

to the strainers.

It is interesting to consider some of the factors involved in purification by this means.

In our Laboratories the data for the following curves were obtained by passing a special used foil rolling oil through a bed of filter clay in a glass column (Fig. 5). Although foil rolling oil was used because of the high ash sample obtainable, the principles involved are believed applicable to any similar product.

Several things are apparent upon inspection of these curves:

1. Long after the filter presumably would have been changed some mechanical filtration still took place, as the ash content of the filtrate is less than that of the oil charged to the column.

2. Such equipment clearly could not be used for mechanical filtration since it permits 0.06% ash to pass through, whereas only 0.005% is desired.

Table I. Additive and ash values; cold rolling oils. (*Factor by which saponification number is multiplied to find percentage of additive.)

	Exhausted Filter			New Filter		
	Before	After	% Removed	Before	After	% Removed
Additive Values	5.6X*	5.5X	1.8	5.9X	5.5X	6.8
Ash Values	0.013	0.012	7.7	0.014	0.008	42.9

3. In this particular case 60 cc's of filtrate were collected before the saponification number began to rise, and 85 cc's came through before this number reached that of the original charge. A simple calculation shows that the 50 gms of clay removed about 3.5 gms of additive. Since each 50 gms of dry clay weighs 90 gms when wet with oil, 36.5 gms of oil and additive are lost representing 33.8 gms of oil and 6.2 gms of additive for each 50 gms of dry clay used. The wetting losses cannot be avoided, but there is evidence that in plant-scale operation some additive initially removed may be desorbed and returned to the system.

4. The desirable zone of adsorptive filtration lies in the region between 85 and 249 cc's in this particular case. In this region no additive loss occurs, yet solid impurities are removed. As a matter of interest an electron micrograph of a similar oil was made, and the particles were found to be remarkably small, some with a diameter in the range of 200 Angstrom units (made by courtesy of The Texas Co.) (Fig. 6). Such particles could not practically be removed by most mechanical filters, but are easily extracted by adsorptive means.

In conclusion, the additive concentrations and ash values before and after passing through new and exhausted filters are shown in Table I. These values indicate what might be expected of fairly high flow rates, and that the ash removal is more rapid than is the depletion of additive.

ACKNOWLEDGMENTS. The assistance of our sheet plants, Metallurgical Research Laboratories, and Staff is acknowledged, as well as Management for permitting the paper to be presented.

Current Literature

Snap-Tite Hi-Flow Coupling. Bulletin 230 describes the new quick-connect/disconnect hi-flow coupling, its uses, and construction; lists complete dimensions of all parts; and gives ordering information for spare parts, as well as for complete couplings. (Snap-Tite, Inc., LE-11/5, Clark Rd., Union City, Pa.)

Type RH Laboratory Heater. Bulletin 627 discusses the features of a new laboratory heater (Type RH) with a built-in rheostat, including range, construction, ease of setting rheostat control, and temperature reproducibility. (Precision Scientific Co., LE-11/5, 3737 W. Cortland St., Chicago 47, Ill.)

Trabon Lubrication Pumps. Four new bulletins, No. 5531 (Double Acting Hydraulic Pumps), No. 5533 (Single Acting Hydraulic Pumps), No. 5535 (Single Acting Pneumatic Pumps), and No. 5537 (Cam Driven Mechanical

(Continued on p. 349)

Integrally Sealed Ball Bearings

by L. D. Cobb*

In this paper the author has endeavored to trace the history of seal bearing design and development from its inception to the current time. While no attempt has been made to outline all current and past seal bearing designs, the basic features common to nearly all existing designs are discussed. The paper also discusses a customer requirement survey which indicated that sealed bearings in the near future must be more efficient, carry heavier loading at higher speeds, and must operate under combined self-generated or ambient temperature conditions exceeding 500 F. Paper also discloses a new seal design and new ball bearing ring and ball material which will fulfill the above-mentioned requirements when suitable mediums are available for proper bearing lubrication.

Integrally sealed ball bearings have been used in ever increasing quantities since about 1928 when the idea of a packaged ball bearing, integral sealing, and lubrication was first conceived. The primary acceptance of this design was simplicity of installing the bearing unit into mechanisms. Also, improved performance was achieved by controlled sealing, use of correct type and volume of grease lubrication, and elimination of bearing contamination during periods of storage or assembly of bearings into mechanisms.

Prior to discussions on future designs past progress should be evaluated, as nearly all manufacturers of sealed ball bearings have followed much the same pattern. No attempt will be made in this paper to list and evaluate all the available commercial seal designs but, rather, to evaluate progress on sealed bearings with which the writer has been specifically concerned as this covers the entire scope fairly well.

As will be noted in Fig. 1, the first integral bearing seal, titled 1928, was composed of a standard ball bearing encased in a brass shell "A" which, in turn, encompassed and positioned a felt sealing member "B" on the shaft shoulder "C" adjacent to bearing proper. This design had two adverse features in that the bearing could not be removed from the shaft without ruining the seal and, also, that seal interference fit could not be accurately controlled. The second design, devised in 1930 and shown in Fig. 1, eliminated these two adverse features by extending the bearing inner ring "D" and riding the seal "B" on this extension. However, this design resulted in non-standard bearing O.D. due to added thickness of enclosing shell "A". Also, the bearing outer ring "E" could, under some conditions, revolve in the brass shell "A" creating wear debris which contaminated the bearing. The third design, evolved in 1934 and shown in Fig. 1, eliminated all the above mentioned objections by supporting the felt and its metal enclosure in the bearing outer ring bore in position "F". This 1934 basic design is still widely used, with minor modifications. One such modification, also shown in Fig. 1, (1936), is to replace the felt member with a free running metal slinger "G" press fitted onto bearing inner ring O.D. This modification was used to lower bearing seal torque and allow higher bearing operating temperatures than those allowed by the felt component.

All of these designs resulted in non-standard bearing dimensions and the next step, evolved about 1944, was to develop a seal for standard single row bearing external geometry which allows complete interchangeability of sealed, shielded, or open bearings. In this 1944 design the sealing member "H" is Buna-N coated cloth fabric held into bearing outer ring with a sheet metal stamping "J" as shown with Buna-N material contacting on inner ring seal notch to form the seal joint. Seals of this type are difficult to control to insure a low pressure contact of flexible seal material to bearing inner ring.

In previous evaluations only the detrimental features have been pointed out, and it should be recognized that for usage desired in the past and on many current applications these seals all give very acceptable operation.

As will be noted, all the seals up to this point were non-removable which is acceptable if bearings are packed with stable greases and are not subjected to extended storage periods of the order of 5 to 10 years prior to usage. Many customers, particularly the Armed Services, are now demanding a seal which can be removed to allow bearing cleaning and regreasing in their storage depots where bearings are shelf-stored, sometimes under adverse storage conditions, for extended periods. While a removable type seal had been designed and put into limited production as early as 1930, it was not until about 1947 that synthetic rubbers were available which would withstand chemical attack of all the lubricants and conditions required in sealed bearings. As these new materials of the butadiene-acrylonitrilene rubber and like compounds opened up new avenues of approach in seal designs, a canvass was made of customer's current and anticipated usage so that with good planning a new sealed bearing design could be developed which would not be obsoleted for many years, as tooling up new designs for the many sizes of bearings required is a very costly affair.

In this canvass the following was found to be required as soon as possible as bearing limitations were holding up mechanism designs currently in

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(This paper was sponsored by the ASLE Technical Committee on Bearings & Bearing Lubrication, and presented at the ASLE 10th Annual Meeting, Chicago, April 14, 1955.)

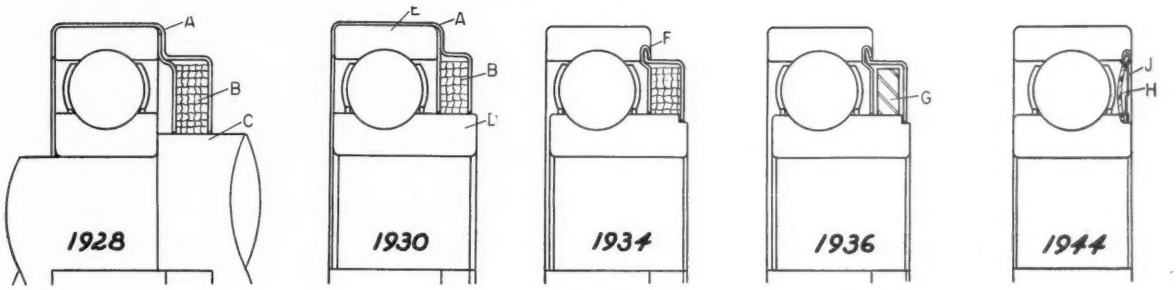

Fig. 1. Basic seal bearing designs.

the planning state:

(a) More efficient seals with lower torque,

(b) Higher bearing speeds of 1,000,000 and higher DN values,

(c) Greater bearing load-carrying ability,

(d) —80 to +500 F. bearing operational temperatures,

(e) Greater bearing life span.

The above has been partially or fully consummated as follows: More efficient sealing indicates that there will be no lubricant leakage or contamination entrance allowed through any seal-race ring junction. Due to space and torque consideration a compromise must be effected or sealing efficiency at location where relative rotation exists, but no compromise need be made at outer ring seal junction. For lowest possible seal torque and maximum seal efficiency, control of sealing pressure against relative moving race ring must be controllably maintained at all times regardless of bearing operating eccentricity, inner or outer race ring rotation, axial bearing component shift due to allowable bearing end play, or bearing operational speed and temperature.

An integral seal bearing as shown in Fig. 2 has been designed, laboratory tested, and is now becoming available, which appears to satisfy these conditions. For discussion purposes this design will be entitled seal "Z". As will be noted this seal "Z" is essentially of molded construction composed of flexible material with metal supporting inserts. The metal insert "A" is a continuous metal ring which places the molded flexible material contacting the outer race seal groove in compression, thus forming a tight joint holding the seal firmly in place and also making an air-tight closure. The seal can easily be removed with a thin-bladed sharp pointed tool and replaced without damage as slight damage to metal ring "A" has no effect on balance of seal. Seal will resist axial applied loads of over 8 to 10 pounds tending to force seal out of bearing on a 203 basic bearing size.

A second insert "B" supports the sealing lip "D" in relative required contact on inner ring race seal groove "E". This insert "B" is used to provide resistance to ballooning of seal with outer ring rotation and to stiffen seal section at "D" to eliminate synthetic rubber material set, thus assuring seal contact with inner ring under all operational conditions. The portion of flexible material "C" forms a fairly constant low rate spring which maintains the pressure of seal on inner ring nearly a constant at all times regardless of positioning of outer race to inner race due to manufacturing or operational misalignment of bearing rings, allowable bearing end play, or reasonable off tolerance seal grooves incurred during manufacturing. Use of this spring "C" reduces variable in seal torque to under 50% of that incurred where one uninterrupted insert is used to support the seal in outer ring groove and support seal lip "D". All this adds up to more uniform torque values, more consistent efficient sealing and, due to lower torque values, higher limiting speeds at lower self-induced operating temperatures are achieved. Also, the seal portion sliding on inner ring seal groove is molded to eliminate dimensional variables and ravel usually synonymous with die trim due to die wear.

Currently, this new type seal is made of butadiene-acrylonitrilene rubber and polyacrylate rubber depending on temperature requirements. The butadiene-acrylonitrilene is a special compound for low coefficient of friction and high resistance to all synthetic or aromatic lubricants within a recommended temperature range of —40 to +225 F. for extended periods, and 300 F. maximum for one (1)

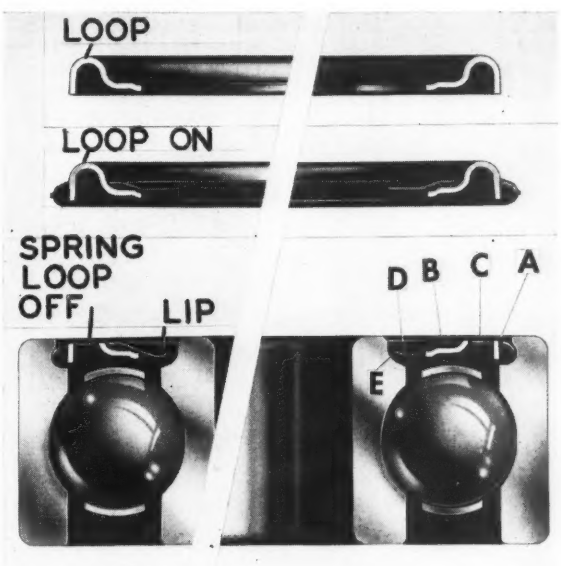

Fig. 2. Z-Type Seal construction. (Top) Metal insert; (Middle) Insert molded in synthetic rubber.

hour. These values are conservative and material has been successfully used at —80 to +250 F. where critical sealing is not a requirement. Butadiene-acrylonitrilene rubber, which will withstand higher temperatures, has been compounded but is not currently recommended because of relatively high cost. Currently, for these higher temperatures, a polyacrylate rubber material which will withstand —40 to +350 F. for continuous operation and 450 F. for limited periods is recommended. This material apparently will withstand all lubricants except the di-ester type which creates excessive swelling at temperatures over 250 F. Many other synthetic compounds which are of flexible material are being considered and it appears that seals of construction shown will be available for temperatures approaching 500 F. on a commercial basis in the near future. Currently, most of these explored high temperature materials have poor set or tear characteristics which are not insurmountable to overcome.

The final acceptance of a seal design, of course, is its functional efficiency against passage of fluids, air or gases, or contaminants. One of the most drastic contaminant tests conceived is to subject the seal to a turbulent atmosphere super-saturated with various fine grades of alundum. Under this test any seal which passes a 100-hour test with bearing inner ring revolving is considered a good seal; if it passes a 1000-hour test it is considered

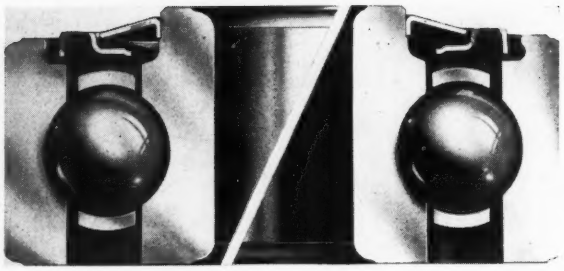

Fig. 4. Combination Z and Slinger Seal.

superior. For fluid testing the bearing seal is subjected to a 6″ head of 00 cup grease with bearing in static and dynamic operational conditions. Bearings passing this drastic 50-hour fluid test are considered good, and after 100 hours, superior. Fig. 3 shows laboratory test results on a basic 201 to 205 size of bearing and shows improved efficiency of design developed since 1928.

If added bearing width is allowable, a metal slinger can be attached to inner ring as shown in Fig. 4. With the slinger attachment all seals passed the 1000-hour acceptance on contaminant test. Fluid sealing showed no improvement with slinger addition, and it is felt that a slinger is not warranted except under very severe contaminant or as a deflector for external oil splash or jet streams.

As noted before, requirements of seal bearings are for greater load-carrying ability, higher permissible speeds, and temperatures of —40 to +500 F. The seals required new designing to obtain lower torque, to withstand higher temperatures, and to improve efficiency; greater load-carrying ability, higher speed, and —40 to 500 F.; also, new improvements on basic bearing parts and their lubrication.

With oil lubrication it is possible to cool the bearing operating parts by controlled inlet oil temperatures. However, in integral sealed, grease lubricated bearings, no cooling medium is available except through housings, shafts, or ventilation, but these mediums in some cases will be specified to operate at approximately 500 F. themselves. Thus, if 500 F. is desired for an average mechanism temperature, the ball bearings will also operate at 500 F. or higher. This re-

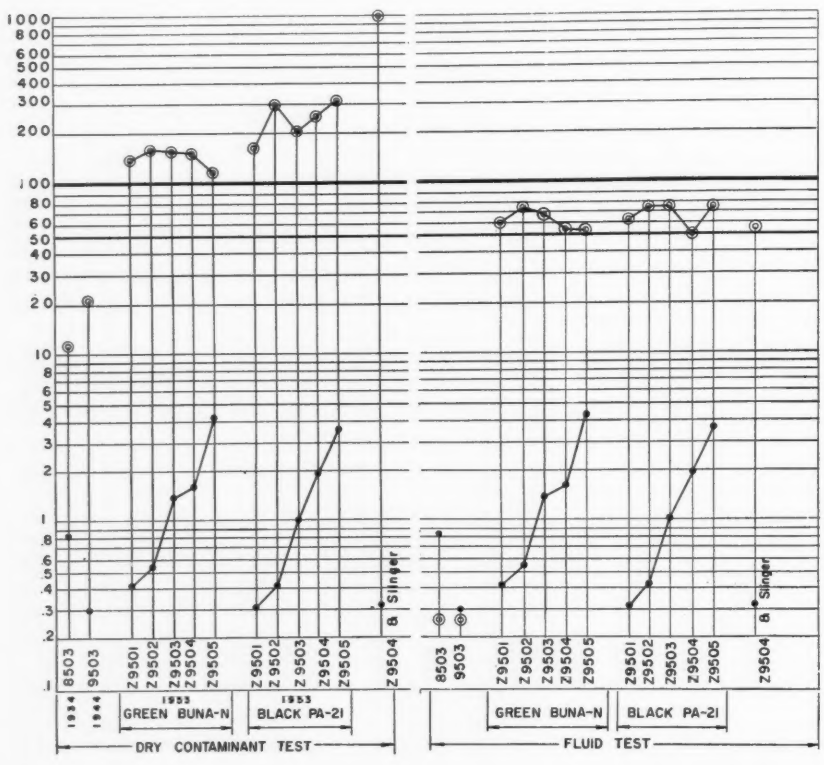

Fig. 3. Relative torque and seal efficiency; contaminant and fluid tests; inner ring velocity, 450 ft/min. Key: ⊙, Average seal life in hours; ●, Torque, oz. inches, one seal, bearing ¼ full, Andok-C, I.R. 30 rpm.

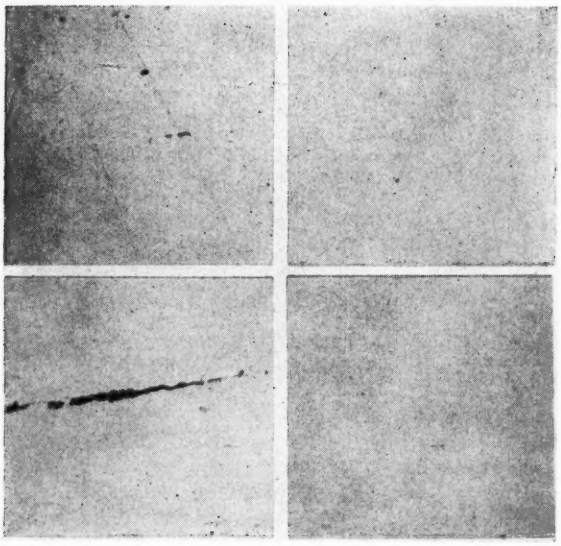

Fig. 5. Inclusion Pattern, vacuum melted (right) vs. standard steel (left). (Top row) 52100 Steel; (Bottom row) Tool Steel.

sults in three possible adverse conditions:

(1) Standard ball and ring material dimensional instability at high temperature,

(2) Metal fatigue capacity reduction due to possible reduction in ball and race hardness,

(3) Inability of lubricant to protect bearing surfaces from cold welding or wear due to operation in a possible non-reducing atmosphere.

Dimensional stability can be improved by drawing the steel, if modified AISI-52100 or modified AISI-51100, at either 360 F. or 50 F. above operating temperature, whichever is the highest, for two 2-hour periods allowing cooling to room temperature in air between heat cycles. The lowest Rockwell hardness recommended, resulting from drawing and operation time and temperature, is about 58 Rc as bearing capacity falls off very rapidly at lower than 58 Rc for ring and ball of modified AISI-52100 or modified AISI-51100 materials. The other alternative is to use a good grade of clean, high temperature tool steel which does not draw appreciably up to 800 to 1000 F. Several non-critical alloy steels are available, processed by vacuum melting, which are acceptable but currently costly. It should be noted that vacuum melting process is one means of obtaining the cleanliness required. Other processing resulting in like material cleanliness should operate equally as well as vacuum melting method. A typical inclusion pattern of standard vs. vacuum melted 52100 and, also, of a tool steel alloy of 1.00% vanadium, .65% carbon, 5.0% chrome, 5% molybdenum, balance iron is shown in Fig. 5 with field at 100 magnifications.

Under standard laboratory test conditions of heavy radial loading, 2000 inner ring RPM, SAE 30 oil jet lubrication, and 130 F. as measured on the bearing outer ring, the vacuum melted 52100 steel allows about twice the average life of bearings made of standard 52100 steel. Under the same conditions except with temperature of 450 F., the semi-critical vacuum melted tool steel allowed the same average life as did the vacuum melted 52100 at 130 F. Prior to vacuum melting the tool steel of the same analysis mentioned above allowed ouly about 25% of the average life obtained with 52100 standard steel under heavy overload conditions at 130 F. described above.

From the previous discussion it can be stated that progress has been made as follows:

Item 1. More efficient sealing commensurate with acceptable bearing seal torque,

Item 2. Seal material for seals in Item 1 with increased operating temperature range approaching 500 F.,

Item 3. Bearing component material tested at 500 F. and useable up to 800 F. without drastic material-physical changes resulting in increased load carrying ability from normal room to 800 F.

The greatest difficulty in obtaining desired integral seal bearing operational characteristics is grease lubrication, as at over 250 F. to 300 F. load carrying ability of available greases as determined by bearing life for given test speed, load, and temperature conditions fall off very rapidly upon approaching these temperatures. It has been found that for temperatures in excess of 200 to 250 F., useful grease life under bearing operating conditions can be extended by increased grease volume, but this does not allow life increases to the degree desired by bearing users. Lubricant research engineers are working on this problem and indications are that suitable lubricants for 500 F., integral seal bearing operation may be available in the not too distant future. When such lubricants are available, integrally sealed bearings capable of desired high speed, heavy overload, —100 to +500 F. operation, and satisfactory life will be available allowing lighter weight, more efficient packaged ball bearings which should open up tremendous new markets.

COMMENTARY, by J. H. Gustafson (Marlin-Rockwell Corp.). Mr. Cobb has presented an interesting history of the development of sealed ball bearings. Most bearing companies are involved in development programs on this type of seal to improve sealing efficiency, permit a higher operating speed, and to develop more satisfactory materials for high temperature operation. For temperatures exceeding 350 F. we have had a certain amount of success with silastic materials, but their inherent low tensile strength and incompatibility with silicone lubricants and exceedingly high cost have limited their wide spread use.

The obvious advantages of a packaged, pre-lubricated ball bearing has resulted in the evolutionary development of a synthetic rubber contact rubbing seal which is now common throughout most of the bearing industry. New Departure has traced the development of their seal through their felt seal type of bearing, while others such as Marlin-Rockwell have developed a similar type of rubber seal from

a single row width metal shielded bearing. It is interesting to note that while the development of this sealed bearing progressed through different channels, the resulting seals are very similar.

We were interested to note Mr. Cobb's comments in regard to testing the sealing efficiency of bearings in the presence of dust and cup grease. We have found that the sealing efficiency requirements of customers vary considerably depending upon the application requirements. A standardized test to check sealing efficiency would probably be of great value to bearing manufacturers and users alike.

The information which Mr. Cobb discloses concerning high temperature alloy steels and vacuum melted steels is of particular interest at this time. The operational requirements for bearings made today, and the projected demands on bearing performance for carrying heavy loads at high speeds and high temperatures, indicate a very great need for additional research on this subject.

We believe that the bearing manufacturers and the equipment designers have outstripped the lubricant manufacturers in their search for materials suitable for high temperature operation. Bearing applications involving temperatures of 800 F. and higher are not uncommon. While bearing materials are available for such high temperature operation, no satisfactory lubricant exists for such applications.

Personals

IN MEMORIAM. Arvid E. Roach (ASLE Director), Supervisor of Bearing Development for General Motors Research Laboratories Div., Detroit, Mich., died July 28 following a long illness.

Born in Philadelphia, Pa., in 1919, Mr. Roach received his Bachelor of Science degree in Electrical Engineering from the University of Delaware in 1941; in 1949 he was awarded a Master of Science degree in Mechanical Engineering from Wayne University where he also served as a special instructor in mechanical engineering. Mr. Roach joined GM Research Labs. in 1940 as a summer student, and as a staff junior engineer in the Administrative Engineering Dept. following graduation from the University of Delaware. From 1944 to 1946 he was an officer in the U. S. Navy, first at St. Juliens Creek, Va., conducting tests in experimental loading of aircraft projectiles; and later at Solomons, Md., as underwater explosives test officer in charge of testing experimental and captured enemy mines, torpedo warheads, and depth charges. He returned to General Motors following Navy service and in 1948 transferred to the Mechanical Development Dept. where he specialized in bearings and lubrication problems. Most of his work was devoted to the fundamentals of friction and wear and the score resistance of bearing materials.

A frequent speaker before engineering and technical societies, Mr. Roach presented papers on lubrication, bearing design and materials, the theory and design of centrifugal pumps, applied hydraulics, and industrial mathematics. Last November with C. L. Goodzeit and R. P. Hunnicutt of GM Research Labs., he reported to ASME a new basis for selecting bearing materials. His theory was deducted from test data with virtually all types of bearing metals rubbing on steel journals. Tests included most metallic elements in bearings, both systematically and on a comparative basis, aimed to determine basically why some metals weld and others do not when they slide or rub against other metal surfaces. More recently Mr. Roach collaborated with P. F. Chenea, Engineering Dean of Purdue University, in developing a new approach to the theory of mechanical wear, based on application of a principle of thermodynamics. They evolved a general equation for the rate of mechanical wear, derived from mass and energy considerations.

Mr. Roach held memberships and was an active committeeman in many engineering and technical societies: he was Chairman of the Lubrication Research Committee of ASME; Chairman of the Technical Committee on Bearings & Bearing Lubrication, ASLE; and President of the Industrial Mathematics Society. He was a member of the Engineering Society of Detroit, the Society of Automotive Engineers, the Scientific Research Society of America, and the honorary engineering and scientific fraternities, Tau Beta Pi and Sigma Xi.

J. J. O'Connor ("Lubrication Engineering" Editorial Committee Member), McGraw-Hill Publishing Co., New York City, has been appointed Managing Editor of "Power" magazine.

Acheson Colloids Co., Port Huron, Mich., has announced the appointment of **F. M. Hunter** as Supervisor of the laboratory phase of their Sales Engineering Program; and **F. Zajac** as Detroit Service Engineer with headquarters at 3000 E. Grand Blvd., Detroit 2, Mich.

E. C. Gits has been named Vice-President in Charge of the Seal Div., Gits Bros. Mfg. Co., Chicago.

E. S. Moyer, formerly Ass't. Chief Engineer of the Cadillac Tank Plant in Cleveland, has been appointed Ass't. Chief Engineer of Moraine Products Div., GMC, Dayton, Ohio. (Editor's Note: One of Mr. Moyer's noteworthy achievements was designing a cooling system for the M-5 light tank built by Cadillac during World War II.)

Houdaille-Hershey of Indi-

(Continued on p. 353)

Patent Abstracts

(Compiled by Ann Burchick, Aluminum Co. of America.)

Aluminum Distearate Composition and Greases Prepared Therefrom, Patent No. 2,702,792 (K. D. Ashley, assignor to American Cyanamid Co.) A grease composition comprising a major proportion of a hydrocarbon lubricating oil and a minor quantity within the range of about 4% to 12% by weight of an aluminum distearate having a substantial content of aluminum distearate anhydride therein.

Separation of Emulsions, Patent No. 2,702,794 (G. W. Kellogg, assignor to General Motors Corp.) A process for resolving an oil-in-water emulsion comprising treating such emulsion with a water-soluble inorganic salt, agitating the mixture, adding thereto a normally non-emulsifiable light oil, subjecting the mixture to a second period of agitation, and subsequently permitting gravity separation until the oil is substantially separated from the water.

Treatment of Oil for Removing Iron, Patent No. 2,703,306 (G. F. Asselin, assignor, by mesne assignments, to Esso Research & Engineering Co.) A method for removing iron from a mineral oil containing same which comprises contacting said oil with an aqueous solution of ammonium thiocyanate, and recovering said contacted mineral oil having a reduced content of iron.

Stabilized Sulfur-Containing Additives for Lubricants, Patent No. 2,703,318 (W. E. Waddey, J. M. Phelan & D. T. Rogers, assignors to Esso Research & Engineering Co.) A stable sulfur-containing organic product comprising a sulfurized member selected from the group consisting of animal and vegetable oils, fats, rosins, and the acid, ester, hydrogenated and lower alkyl derivatives thereof, said product being stabilized by cooking after sulfurization with 0.02 to 1 part by weight, based on the sulfurized material, of an active olefinic hydrocarbon at a temperature within the range of 60 to 400 F. for one-fourth to 20 hours.

Process for the Clarification of Lubricating Oil Additive Concentrates, Patent No. 2,703,783 (A. H. Popkin, assignor to Esso Research & Engineering Co.) A process for the clarification of oil concentrates of a polymeric lubricating oil additive material selected from the class of polymers of iso-butylene and copolymers of isoprene, said oil concentrates being clouded due to the presence of a material of the class consisting of zinc stearate and carbon black, which comprises heating the oil concentrate to a temperature within a range of from about 210 to about 350 F., subjecting the heated concentrate to the action of steam for from about 5 minutes to about 1 hour and filtering said heated steamed concentrate to obtain a clear, unclouded solution.

Soluble Compositions Containing a 2,5-Dimercapto-1, 3,4,-Thiadiazole Derivative, Patent No. 2,703,785 (E. N. Roberts, E. K. Fields & J. S. Brown, assignors to Standard Oil Co.) An emulsifiable soluble oil composition comprising a major proportion of a hydrocarbon oil, an emulsifying agent, and from about 0.5% to about 10% of the oil-soluble reaction product of an aldehyde, a mercaptan selected from the class consisting of an aliphatic mercaptan and an aromatic mercaptan, and 2,5-dimercapto-1, 3, 4-thiadiazole, said aldehyde, mercaptan and 2,5-dimercapto-1, 3, 4-thiadiazole being reacted in the molar ratio of from 2:2:1 to 10:2:9.

Lubricating Composition, Patent No. 2,703,786 (D. W. Young & D. L. Cottle, assignors to Esso Research & Engineering Co.) An improved lubricating oil composition having outstanding resistance to oxidation which comprises a mineral lubricating oil containing combined therein from 0.2% to 10.0% by weight of an oil soluble metal derivative of an alkyl phenol sulfide.

Process for the Preparation of Lubricating Oil Additives, Patent No. 2,703, 817 (G. E. Serniuk, assignor to Esso Research & Engineering Co.) An improved process for the manufacture of a lubricating oil additive material having the desirable characteristic of depressing the pour point of lubricating oils into which it is incorporated which comprises admixing a vinyl aromatic hydrocarbon compound and a haloalkane in a polar solvent adding thereto a complex obtained by admixing a Friedel-Crafts catalyst with a saturated organic monobasic carboxylic acid chloride having from 2 to 24 carbon atoms, maintaining the temperature of the mixture at one within the range of from —20 C. to about 50 C. for a period of time sufficient to polymerize substantially all of the vinyl aromatic, and thereafter raising the reaction temperature to one within a range of from 60 C. to 100 C. to complete the reaction.

Thienylthio-Substituted Ethers and Mineral Oil Compositions Containing the Same, Patent No. 2,704,275 (J. W. Brooks, assignor to Socony-Vacuum Oil Co., Inc.) A mineral lubricating oil containing a minor proportion, sufficient to stabilize said oil against oxidation.

Lubricating Composition, Patent No. 2,704,276 (P. R. McCarthy & T. R. Orem, assignors to Gulf Research & Development Co.) A lubricant comprising a dispersion of a compound of a bentonite and an organic base containing 10 to 20 carbon atoms in a solution of an oil and a polymer selected from the group consisting of polymerized ethylene, polymerized esters of acrylic acid, and polymrized esters of alpha-methacrylic acid, wherein the amount of said bentonite compound is sufficient to produce a composition having the consistency of a grease and wherein the amount of said polymer is sufficient to improve the water-leaching characteristic of said lubricant.

Mineral Oil Compositions Containing Esterified Copolymers of Alpha-Beta-Unsaturated Polybasic Acids with Allyl & Vinyl Ethers, Patent No. 2,704,277 (J. J. Giammarie, assignor to Socony-Vacuum Oil Co., Inc.) A mineral lubricating oil containing a minor proportion, sufficient to lower the pour point of said oil, of an esterified copolymer formed by copolymerizing (1) an ether selected from the group consisting of alkyl vinyl ethers and alkyl allyl ethers, the alkyl portion of said alkyl vinyl being an alkyl radical containing up to and including four carbon atoms and the alkyl portion of said alkyl allyl ether being a primary, normal, alkyl radical having from 1 to about 18 carbon atoms, with (2) an alpha, beta-unsaturated polycarboxylic acid material selected from the group consisting of maleic acid, fumaric acid, itaconic acid, glutaconic acid, mesacpnic acid, citraconic acid, aconitic acid and the anhydrides of said acids, and esterifying the copolymer thus obtained with (3) an alcohol material, said alcohol material being selected from the group consisting of tetradecyl alcohol and a mixture of primary, normal, saturated alcohols having an average of about 14 carbon atoms per molecule when the ether utilized in preparing the copolymer is an alkyl vinyl ether and being a primary, normal, saturated alcohol having from 1 to about 18 carbon atoms when the ether utilized in preparing the copolymer is an alkyl allyl ether.

Wire Drawing Composition, Patent No. 2,740,744 (G. L. Hugel, M. Lerer & R. J. Marie Courtel, assignors to Institut Francais du Petrole des Carburants et Lubrifiants, Paris, France.) A wire drawing paste composed of 44 parts of talc powder, 54 parts of 800 M. W. polyethylene glycol and 2 parts of p-dimethoxy-diphenyl-trichlorethane.

Stabilized Extreme Pressure Lubricant, Patent No. 2,704,745 (H. L. Kopp & L. W. Manley, assignors to Socony-Vacuum Oil Co., Inc.) A lubricant composition comprising a major proportion of a refined mineral lubricating oil and (1) from about 0.1% to about 3.0% by weight of a sulfur- and nitrogen-containing organic compound selected from the group consisting of 2-benzothiazyl N-N diethyl thio carbamyl sulfide, 2-benzothiazyl N-N dibutyl thio carbamyl sulfide, oxypenthamethylene thio carbamyl diethyl thio carbamyl sulfide, bis-4-methybenzothiazyl disulfide,

(Continued on p. 357)

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(Men of Lubrication, from p. 305)

ball in a cup having a slightly greater radius than the ball. A small quantity of oil is placed in the cup and then the ball is dropped in place. A clearance between ball and cup is maintained by three small projections on the cup. The cup is then inverted and the time taken for the ball to fall free is observed. This time interval is proportional to the viscosity. Michell apparently arrived at the proportions by an approximate calculation based on the time for a circular disc to separate from a flat plate when there is an intervening fluid film. However, the uncertainties of manufacture make calculations of only qualitative value and each instrument is calibrated. The device does not appear to be widely used in this country, but it would seem that it could be of considerable practical value.

In 1934 Michell was elected a fellow of the Royal Society of London, and in 1938 he received the Kernot Medal from the University of Melbourne. He was awarded the James Watt International Medal by the Institution of Mechanical Engineers in 1943. His comprehensive book, "Lubrication," was published in 1950. It contains a very thorough treatment of the theory with much information not to be found elsewhere.

Michell was one of a family of five children, three of which were girls. His elder brother, John Henry Michell, is the J. H. Michell who contributed so prolifically to the theory of elasticity and hydrodynamics around the turn of the century. J. H. Michell died in 1940 and a very fine outline of his life and work was written by his brother, A. G. M. Michell, for the Obituary Notices of Fellows of the Royal Society.

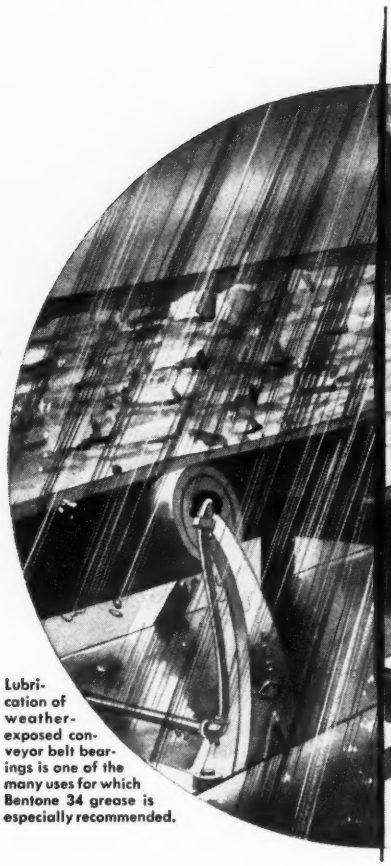
A. G. M. Michell spent some time in the United States during the nineteen twenties in connection with his crankless engine and he became quite well known in American engineering circles. He is said to be of a retiring nature and to find his relaxation in horticulture around his country home in Bunyip Victoria.

ACKNOWLEDGEMENTS. The main sources consulted in the preparation of this article have been a bulletin of the Institution of Mechanical Engineers on the occasion of the award of the James Watt International Medal to Mr. A. G. M. Michell, F. R. S.; and Michell's papers and book, "Lubrication."

(This is the fifth in a series of articles by F. R. Archibald, Analyst, of Arthur D. Little, Inc.)

(Lube Lines, from p. 293)

Wrapping papers impregnated or coated with the inhibitor are available commercially and represent the most common form of application. The dry inhibitors may also be applied by spraying, flocking, or dusting. Trays of inhibitor are also sometimes used for large spaces. If a system to be protected must be vented to the air, protection can be achieved by placing the inhibitor in the breather pipes so that air entering the chamber passes over the crys-



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(Lube Lines, from p. 346)
tals and maintains saturation.

Many volatile organic amine salts are known to inhibit corrosion of ferrous metals, and some of these have been patented as volatile corrosion inhibitors. Some are available commercially as solids. For special applications some of these materials may be dissolved or suspended in water, alcohol, or oil, and applied directly to the surfaces to be protected or to contiguous areas.

The value of paper and crystals currently sold for vapor phase protection is $2,000,000 to $3,000,000 per year. Several engine manufacturers and aircraft engine rebuilders make extensive use of the volatile corrosion inhibitor method of rust protection for packaging spare parts. It is expected that this method will take over a large share of the rust preventive market. (Refs.: "Halt Rust in Parts Shipments," by R. C. Lambrecht, Factory Management & Maintenance, March 1953; and "Army Ordnance Corps Develops Special Spray Gun to Aid in Rust Protection," Anon., Automotive Industries, Jan. 15, 1955.)

(Book Reviews, from p. 336)
tants in advance, this new industrial guide booklet reveals that incidence of industrial dermatitis ranges from a high of 7.5% in the synthetic resin manufacturing industry, to a low of 0.4% for automobile manufacture; presents a brief study of the anatomy and physiology of the skin; and lists causes of occupational skin diseases under five main headings: Mechanical (friction, pressure, injury), Physical (heat, cold, radiation), Plant Poisons (several hundred woods and plants), Biological (bacteria, fungi, mites, etc.), and Chemical Agents ("primary irritants," such as strong acids or other corrosives; and "sensitizers" like TNT or resins that do not cause dermatitis at once but pave the way on further contact).

Measures for preventing dermatoses on the job are outlined, chief among these being personal cleanliness. An effective program of personal hygiene will provide for quick removal from the skin of any irritants unavoidably contacted at work; such a program should: (1) Provide running hot and cold water, a proper industrial cleanser, and towels no further than 100 feet from the place of work, and enough showers so no worker need wait more than a few minutes; (2) Educate workers thoroughly in the importance of personal cleanliness through posters, safety meetings, and campaigns; (3) Allow for inspection of workers' skin at periodic intervals; and (4) Provide the skin cleaner best fitted to combat the

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irritants involved in a particular type of industry.

Beyond personal cleanliness, "environmental cleanliness" (often called "good housekeeping") is essential in any industrial plant. Good industrial housekeeping reaches in many corners, and includes proper plant ventilation and design in order to reduce initial contact with irritants to a minimum, and the provision of protective clothing where warranted. Of considerable help, too, are vanishing, water-repellent, or solvent-repellent ointments and creams, and special lotions to prevent excessive drying and chapping of the skin.

Difficult to detect and diagnose, often resistant to treatment, most industrial dermatoses have proved to be test cases of the old adage, "an ounce of prevention is worth a pound of cure." This booklet is, in effect, the first industry-wide publication of these occupational health hazards compiled for the instruction of industrial management.

(Current Literature, from p. 339)

Pumps), give a detailed description of the design features, with diagrams and operating specifications, of each of a particular series of Trabon lubrication pumps. The oil and grease reservoir assemblies are illustrated, and dimensions and capacities are tabulated. (Trabon Engineering Corp., LE-11/5, 1814 E. 40th St., Cleveland 3, Ohio.)

"A List of 'dag' Dispersions for Industry," a revised new booklet, lists 44 colloidal and semi-colloidal dispersions for operational functions, maintenance, lubrication, machine design, and other industrial applications. These products include dispersions of graphite, molybdenum disulfide, mica, vermiculite, zinc oxide, and acetylene black; carriers and diluents are given for each product, along with typical applications and important physical data. Eight entirely new dispersions include such varied products as a corrosion-resistant coating for

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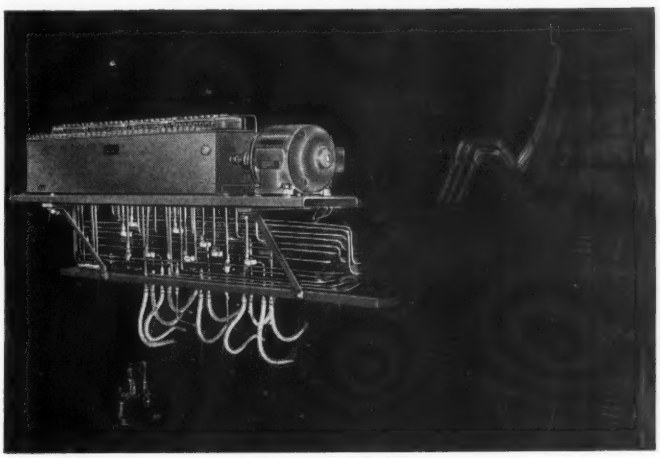

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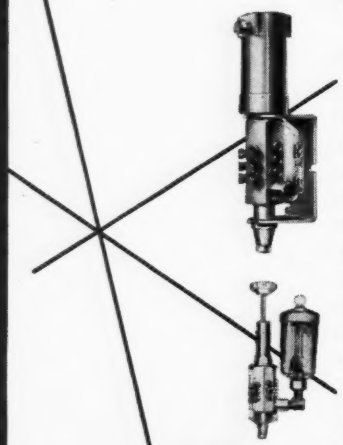

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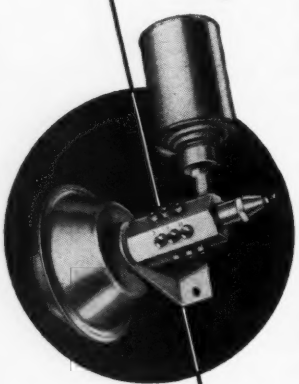

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PIONEER BUILDERS OF LUBRICATING EQUIPMENT FOR A QUARTER CENTURY

dry-film lubrication, an anti-seize compound for oxygen systems, and an improved interior wall coating for cathode-ray tubes. (Acheson Colloids Co., LE-11/5, 1630 Washington Ave., Port Huron, Mich.)

"Modern Filtration," a new catalog of sintered bronze filters (the relatively new filtering medium made by fusing millions of bronze balls into a porous element that never needs replacing and is unaffected by extreme temperatures and most corrosive fluids and greases), includes basic information on the advantages of porous bronze as a filtering medium, and engineering data on bronze filters already in production for aircraft, chemical, petroleum, and other industrial applications. (Permanent Filter Corp., LE-11/5, 1800 W. Washington Blvd., Los Angeles, Calif.)

"The Molykote Story" describes the pioneering in lubrication's "fringe area" which led to the development of the Molykote line of molybdenum disulfide lubricants. Of particular interest are the research facilities and equipment maintained at the company's Basic Research & Development Laboratory in Munich, Germany, as well as those of its Stamford, Conn., plant. (The Alpha Molykote Corp., LE-11/5, 65 Harvard Ave., Stamford, Conn.)

Electroflo Valve. Bulletin 210, complete with flow charts, describes Electroflo, an electrically operated valve for controlling large flow volume. (Hays Manufacturing Co., LE-11/5, Dept. 617, W. 12th St., Erie, Pa.)

"Permatex Industrial Sealant Guide," a new reference guide and chart, describes the proper sealants for use in hundreds of industrial applications including industrial refrigeration, plumbing and heating, exposed and underground pipe lines and tanks, machine tooling equipment, oxygen assemblies, and all mechanical and industrial equipment requiring leakproof air, water and vapor, and chemical connections. Also included are descriptions of seal-

ants exceptionally resistant to vibration; to hot and cold acids, alkali, and salt solutions; and to coolant chemicals. (Permatex Co., Inc., LE-11/5, P. O. Box 1, Brooklyn 25, N. Y.)

Silastic, the Dow Corning silicone rubber, is briefed in a new brochure heavily illustrated with charts, tables, graphs, and actual application photographs. Divided into five sections, it details the properties and performance of Silastic as affected by extreme temperatures, weathering, compression, chemicals, and dielectric service. (Dow Corning Corp., LE-11/5, Midland, Mich.)

(Lube Abstracts, from p. 312)

study of causes underlying the greater strain-hardening effects observed in the deformation of metals in surface-active lubricants when compared with inert lubricants. Active lubricants used: 0.1% stearic acid solns. in nonpolar kerosene; 0.2% oleic acid soln. in pure petrolatum oil. Metals used in experiments: zinc and copper.

New Principle for Studying the Effectiveness of Lubricants in the Working of Metals by Pressure, by S. Ya Veiler & L. A. Shreiner; "Doklady Akademii Nauk SSSR," Vol. 68, 1949, No. 2, pp. 325-328; 3 figs., 1 table, 1,150 words (Order No. 2430, Price $2.10). Description of a simple and convenient method developed by authors for quantitative appraisal of lubricating value of lubricants in the pressure working of metals. Test data on variation of force required in different successive passes without the use of a lubricant. Data on variation of maximum force required as a function of passes, with and without lubricant, based on laboratory experiments and plant trials. Merits of proposed method for use at the shop.

(Section News, from p. 324)

Today on Fire-Resistant Turbine Fluids."

1955-1956 Program. **November 10,** J. T. Wilson, The Louis Allis Co., "The Effects of Atmospheric Contaminants Upon Grease & Oil Lubrication Applications." **December 8,** Dr. E. K. Gatcombe, U. S. Naval Post Graduate School, "A Review of Some Hydrodynamic Lubrication Applications." **January 12, 1956,** L. D. Cobb, New Departure Div., GMC, "New Departure Ball Bearing Research Activities." **February 9,** H. P. Henderson, Worthington Corp., "Lubrication Problems of Gas & Air Compressors." **March 8,** J. P. Critchlow, Gulf Oil Corp., "Gear Lubrication in the Steel Industry." **April 12,** Student Night, with the presentation of the winning student paper in the ASLE N. California Section "Student Paper Contest." **May 10,** Dr. D. F. Wilcock, General Electric Co., "The How & Why of Ball Bearing Lubrication." **June,** Fourth Annual Dinner-Dance. (Submitted by A. C. West, Sec'y.)

TWIN CITIES, June. Election of Officers (see ASLE Directory), followed by an address by D. M. Cleaveland, Bendix Aviation Corp., entitled "Management's Stake in Lubrication." (Submitted by C. D. Johnson, Sec'y.)

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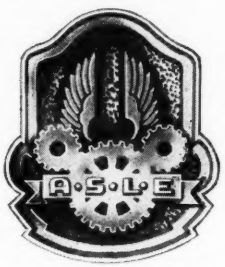

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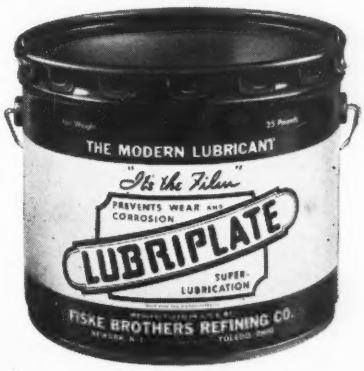

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(Personals, from p. 344)

ana, Inc., Lebanon, Ind., has announced the election of **C. J. Wallace** as Vice-President and General Sales Manager; and the appointment of **J. L. Montmarquet** as Export Manager with headquarters in Ridgefield, N. J.

A. E. Amorosi, formerly with R. M. Hollingshead Corp., has been appointed Chief Chemist with the Alox Corp., Niagara Falls, N. Y.

J. J. Jenkins has been appointed Sales Manager of the Industrial Div. of Lincoln Engineering Co., St. Louis, Mo.

(New Products, from p. 331)

Lincoln Filler Pump. Designed to handle medium as well as light greases, a new low-price Filler Pump can be attached to any 25 lb. to 50 lb. original refinery container by means of three thumb screws. Manufacturer states pump will fill a one-pound capacity grease in only 30 seconds while completely eliminating contamination, air-pockets, and the disagreeable mess of hand-filling methods. Easy pumping action provides efficient, low-cost one-man lubrication rig when used in combination with a filler-type grease gun. Follower plate assures positive prime in any weather and completely empties lubricant container; a self-closing filler socket prevents pumping of lubricant unless filler nipple of gun is inserted in filler socket. Machined piston with solid cork piston ring insures maximum lubricant output on every stroke. (Lincoln Engrg. Co., LE-11/5, 5743 Natural Bridge Ave., St. Louis 20, Mo.)

Circle Seal "200-I Series." A newly designed 3000 psi Circle Seal check valve, designated the "200-I Series," features a heavier, more rugged construction which will absorb more abuse in handling and installation. Location of the static O-ring body seal confines pressure to the strongest part of the body and, along with the heavier construction, permits great versatility in use. Dead

Norgren Spray-Lube System as applied by Square D Co.

to multiple tapping operation, speeds production 30%

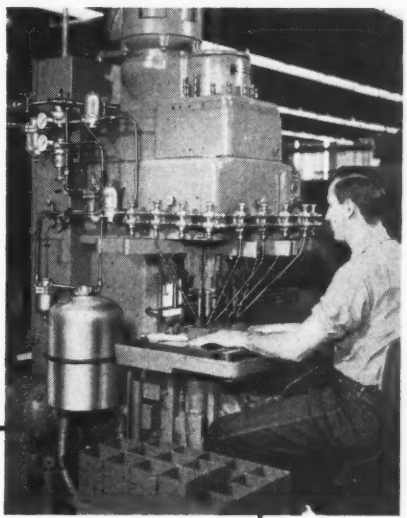
and in addition:

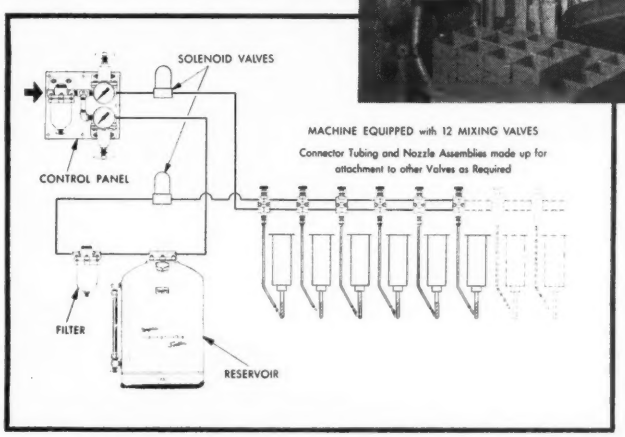
eliminates time and labor formerly required for draining and wiping tapped parts

relieves one man for more productive work

elimination of flood system greatly improves working conditions

substantial saving in lubricant




SOLENOID VALVES
MACHINE EQUIPPED with 12 MIXING VALVES
Connector Tubing and Nozzle Assemblies made up for attachment to other Valves as Required
CONTROL PANEL
FILTER
RESERVOIR

This tapping machine in the Square D Company production line handles a wide variety of tapping operations requiring as many as 12 taps in simultaneous operation. On job shown, tapping 3 holes in each of 2 switchbox covers, Norgren Spray-Lube System increased production 30% . . . eliminated the cleaning process. Even greater increases were reported on other jobs.

The unpleasant and hazardous disadvantages of the previous flood system, splashing lubricant over the operator and the surrounding floor, has been eliminated by Spray-Lube. Lubricant saving (Stanicut No. 107) has been substantial. Solenoid valves permit the Spray-Lube System to be actuated only when taps are approaching and contacting the work.

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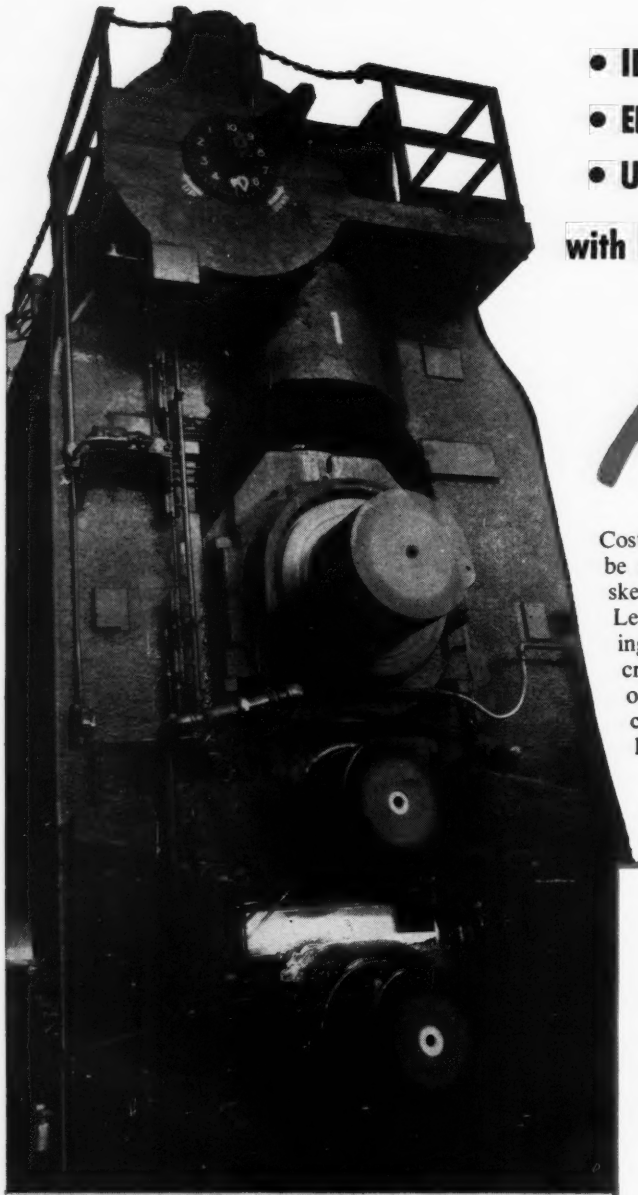

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Honan-Crane Bulk-Type Filters. A new line of bulk refill oil filters has been introduced that offers low-cost, depth-type filtration for applications where oil must be kept absolutely clean. Available in 9 different sizes to meet the filtration needs of the smallest machine tool as well as the largest central system, a dolly-mounted portable unit is also available for general service throughout the plant. An important innovation is a quick opening cover that greatly simplifies changing of refills; covers are fastened by swing bolts which, when loosened, swing down and release cover. On smaller models, cover can be removed by one man; on larger models, a special cover lifting device raises and swings cover to one side allowing unrestricted access to refills. Designed to operate on a by-pass or continuous recirculation basis, the filters employ interchangeable bulk refills of Cranite or Palconia to filter a wide range of industrial oils. (Houdaille-Hershey of Ind., Inc., LE-11/5, 818 Wabash Ave., Lebanon, Ind.)

M-H "Series 400" Valves. A new line of diaphragm control valves, designated "Series 400," incorporates a number of engineering design improvements to provide improved flow characteristics for medium-duty valves. The line includes double-seated, single-seated, and three-way types for use in either "on-off" or proportional control systems. The "on-off" single-seated model can be supplied with a plug having a soft composition disc which provides bubble-tight shutoff; the three-way valve may be used for either diverting or mixing service. Component parts in the series have been standardized to reduce spare part requirements; the compact design of the valves makes them especially suitable for installations in limited space areas. (Minneapolis-Honeywell Regulator Co., LE-11/5, 4494 Wayne Ave., Phila. 44, Pa.)

Purolator Hydraulic Filter. Designed for use on skin-planers, spar profilers, and other large hydraulically-operated machine tools, the new No. 56670 hydraulic filter is capable of filtering micronic-size particles from hydraulic fluids at operating prestures of 1000 psi, and handling flow rates up to 75 gpm. Made of resin-impregnated paper, the miconic filter element is easily replaceable by removing the snap ring holding the case cover in place. The case itself is of all-steel construction and contains an O-ring seal in its head; at either end of the case are 1½" connections. (Purlator Products, Inc., LE-11/5, Rahway, N. J.)

"Revere F-8110" Meter. A new direct-reading, flow-rate meter is announced that provides visual flow-rate indication in circulating, lubricating, and fueling systems. Designed to cover a flow range of 7 to 70 GPH, and to operate at temperatures of — 65 to + 160 F., the unit has only one moving part, is accurate to within ± 2% of the specified flow rate, and has a low-pressure drop. (Revere Corp. of America, LE-11/5, 1000 N. Colony Rd., Wallingford 5, Conn.)

Engineer's Sound Detector. A new, improved listening device makes it possible to detect the smallest internal defects in moving machinery without shutting down or otherwise taking the equipment out of service. Named the "Engineer's Sound Detector," it resembles a doctor's stethoscope in design, and utilizes dual detectors to transmit sounds to one or

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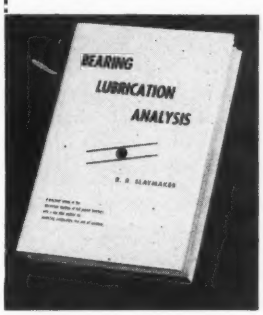

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Letters to the Editor

C. H. Ruth, Jr., writes: I read the article entitled "Atoms Trace The Wear" (Lubrication Engineering, July-August, 1955, p. 238) with considerable interest as I have read many similar articles before.

On p. 241 reference is made to the "Relative Importance of Corrosive, Abrasive, and Friction Wear." Turning back to p. 239 to the paragraph captioned "Corrosive Engine Wear," I note "Various investigators experienced high wear when engines were operated with cold jackets at low speed and mild load."

I realize that the purpose of this article was to impart information on the use of radio-active tracer techniques for measuring engine wear. In reading the article I was not impressed with the conclusion reached to date, none of which to me appeared to be new. If three-quarters of the total wear in these test engines was due to corrosive wear, why do people in the oil industry keep repeating themselves? Anyone interested in this subject learned long ago what causes corrosive engine wear. Now that all are familiar with the disease I think it would be appropriate for these researchers to propose a practical cure. There is a practical cure, but as a fleet operator it was necessary to work out our own cure, and the time has come when no more articles are necessary regarding the cause; let's get a practical answer to the cure and one installed at the level of the basic manufacturer.

Coming Events

OCTOBER, 17-21, American Welding Society (Fall Technical Meeting), Bellevue-Stratford Hotel, Phila., Pa.

17-21, American Society for Metals (National Metal Congress & Exposition), Benjamin Franklin Hotel, Phila., Pa.

17-21, National Safety Council (43rd National Safety Congress & Exposition), Conrad Hilton Hotel, Chicago, Ill.

19-20, American Society of Mechanical Engineers (Fuels-AIME Coal Conference), Neil House, Columbus, Ohio.

24-26, American Standards Association (6th National Conference on Standards), Sheraton Park Hotel, Washington, D. C.

24-28, American Society of Civil Engineers (Annual Convention), Statler Hotel, New York, N. Y.

31 thru Nov. 2, National Lubricating Grease Institute (23rd Annual Meeting), Edgewater Beach Hotel, Chicago, Ill.

31 thru Nov. 2, Society of Automotive Engineers (50th Trans. Meeting), The Chase Hotel, St. Louis, Mo.

NOVEMBER, 2-4, Society of Automotive Engineers (50th Diesel Engine Meeting), The Chase Hotel, St. Louis, Mo.

3-4, Society for Advancement of Management (11th Annual Conference), Statler Hotel, New York, N. Y.

9-10, Society of Automotive Engineers (50th Fuels & Lubricants Meeting), Bellevue-Stratford Hotel, Phila., Pa.

9-12, Society of Naval Architects & Marine Engineers (63rd Annual Meeting), Waldorf Astoria Hotel, New York,, N. Y.

13-18, American Society of Mechanical Engineers (Diamond Jubilee Annual Meeting), Conrad Hilton & Sheraton-Blackstone Hotels, Chicago, Ill.

14-17, American Petroleum Institute (35th Annual Meeting), San Francisco, Calif.

17-18, American Society for Quality Control (10th Mid-West Conference), Schroeder Hotel, Milwaukee, Wisc.

27-30, American Institute of Chemical Engineers (Annual Meeting), Statler Hotel, Detroit, Mich.

27 thru Dec. 1, American Society of Refrigerating Engineers (51st Annual Meeting), Statler Hotel, New York, N. Y.

DECEMBER, 17, Institute of Aeronautical Sciences (Wright Brothers Lecture), Aud. Chamber of Comm. Bldg., Washington, D. C.

(Patent abstracts, from p. 345)

benzothiazyl-N-butyl disulfide and benzothiazyl-tert-butyl disulfide and (2) from about 1% to about 20% by weight of lead naphthenate.

Composition Containing Surface-Esterified Siliceous Solid and Silicone Oil, Patent No. 2, 705,700 (R. K. Iler, assignor to E. I. duPont de Nemours & Co.) A composition comprising an estersil and a silicone oil, said estersil being a supercolloidal substrate coated with—OR groups, the substrate having a surface of silica and having a specific surface area of from 1 to 900 square meters per gram, the coating of—OR groups being chemically bound to said silica, R being a hydrocarbon radical of from 2 to 18 carbon atoms wherein the carbon atom attached to oxygen is also attached to hydrogen, the estersil being organophilic in that it is preferentially wetted by butanol in a butanol-water mixture, the proportion of silicone oil in the composition being, by weight, at least 1 part per 100 parts of estersil for each 100 square meters of surface area per gram of estersil, and the estersil being present in an amount at least sufficient to increase the viscosity of the composition.

Process for Raising the Softening Point of Hydrocarbon Resins, Patent No. 2,705,703 (J. F. McKay, Jr. & D. F. Koenecke, assignors to Esso Research & Engineering Co.) A process for raising the softening point of a petroleum cracked distillate resin which comprises mixing 100 parts of the resin with at least 5 parts of an oily diolefin hydrocarbon polymer and heating the mixture at a temperature of beween 240 and 280 C.

Cutting Oil Derived from Oxidized Petroleum Oil, Patent No. 2,706,176 (D. Frazier, assignor to The Standard Oil Co.) A method which comprises reacting, by admixing while stirring, sulfur chloride and a highly refined solvent extracted mineral oil having an initial viscosity between 75 and 600 SSU at 100 F., an ASTM aniline point of over 105 C., and an aromatic content of below 12%, said solvent-extracted mineral oil having been oxidized prior to admixture with sulfur chloride, to a saponification number between about 20 and 80 and to a neutrali-

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zation number between about 12 and 25, and adding a neutralizing agent to neutralize the reaction product so produced, the proportion of sulfur chloride admixed with the oxidized oil being appreciable but insufficient to add more than about 9.5 parts by weight sulfur to 100 parts by weight oxidized oil.

Method of Inhibiting Corrosion of Metals, Patent No. 2,706714 (W. B. Hughes & R. E. Lembcke, assignors to Cities Service Research & Development Co.) As a new composition of matter, a compound formed by heating together two mols of acetone and one mol of quinoline at reflux temperature and discontinuing the heating when the temperature of the reaction mixture has risen to about 85 C.

Grease Composition, Patent No. 2,706,-715 (R. C. Conner, assignor to The M. W. Kellogg Co.) A grease composition comprising a homegenous admixture of a normally liquid polymer of trifluorochloroethylene boiling within the range of about 150 F. and about 450 F. at 1 millimeter of mercury absolute pressure, and a relatively high melting point waxy polymer of trifluorochloroethylene melting at a temperature above about 350 F., said grease composition having a melting point of approximately 230 F. and a boiling point of approximately 390 F.

Vinyl Ether Containing Hydraulic Fluids, Patent No. 2,706,717 (L. Jakob, assignor to Badische Anilin- & Soda-Fabrik Aktiengesellschaft.) A hydraulic fluid comprising a mixture of between five-sixths and two-thirds by weight of an aliphatic alcohol, selected from the group consisting of fully saturated aliphatic alcohols containing at most 6 carbon atoms and of monoethers of such alcohols containing up to 4 carbon atoms and glycols containing up to 4 carbon atoms, with between one-sixth and one-third by weight of a polymerized vinyl methyl ether of low viscosity, the 15 per cent solution of which in benzene has a K-value of between about 7 and 10.

Stabilization of Organic Sulfur-Containing Compounds, Patent No. 2,706,-177 (T. S. Tutwiler, F. B. Fischl & E. B. Cyphers, assignors to Esso Research & Engineering Co.) A mineral oil composition comprising a major proportion of a mineral oil, an organic sulfur-containing material which normally evolves hydrogen sulfide, and in the range of about 0.1% to 20% by weight, based on the sulfur-containing compound, of an organic ether containing at least one olefinic double bond.

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Ch. Engr. (DA-8160) 48 ME; 3 yrs. principal engr. des. & dev. of eqpt. flow sheets & specs. of utility & power surveys; 9 yrs. project work on pilot plts.; 7 yrs. job engr. pressure vessels, furnaces, piping & structural; 3 yrs. insp. high pressure oil cracking eqpt; US; $13,000.

Plt. Engr. (DA-8501) 33 ME; 2 yrs. asst. supt. maint. chem. plt.; 5 yrs. asst. maint. supt. refinery; has handled cost control corrective & preventative; $8,500.

Plt. Engr. (DA-8504) 28 CE; 1½ yrs. suprv. eqpt. insp. for refinery; 2 yrs. maint. & constr. div.; 3 yrs. preparing mech. specs., invest. des. problems; NY or Ohio; $800.

Designer (305-LE) 30 ME; 4 yrs. proj. engr. resp. for dev. des. of devices relative to temperature control field; some supv. of model shop & test lab.; Midwest; $550.

Ch. Engr. (306-LE) 32 ME; 3 yrs. des. of petro-chem. process plts.; 3½ yrs. res. des. & dev. of fuel oil, & thermodynamic systems; 1½ yrs. des. dev. & tooling for mfgr. of farm eqpt.; $7,500.

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Draftsman (317-LE) 25; BS-Product Des.; 17 mos. layout & des. of optical eqpt.; 14 mos. layout & des. of farm implement including testing experimental models; Chicago; $5,200.

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C-3592 Statistical Quality Control. Chem., Chem. Engr., or Science; age: 25-40; 2 plus yrs. exp. in quality control work in light industry either processing or textiles; know: statistical control procedures; duties: handling statistical quality control in finished goods department or raw materials testing & inspection department; for mfr. of fixtures; sal: $4,200-$5,500; loc: Chicago; empl. will neg. fee.

C-3583 Product Materials Mgr. BS or MS in Met. or Chem.; age: 40-50; 10 plus yrs. exp. in organic chem., organic sealing matls., fluxes, solders & special lubricants; know: polymer coatings for appl. to sheet steel as protective or decorative coatings; duties: report to Dir. of Prod. Res. Dept., resp. for organizing & dir. res. activities on various types of metals potential to mfg. of metal containers; work closely with suppliers of steel, paint & varnish & chem. mfgrers. in dev. formulation of new materials; sal.: $12-15,000; loc: Chicago; Empl. will pay fee.

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C-3539 Lubr. Engr. Pet. Engr. or ME; age 25-34; 3 plus yrs. exp. in plant lubr. work; duties: supervising oilers, specify tests & see that proper lubr. are used; schedule oil changes, order lubricants, & recommend improvements; for Mfr. of Al. & Chem; sal: $480 to $610/mo; loc: Ohio.

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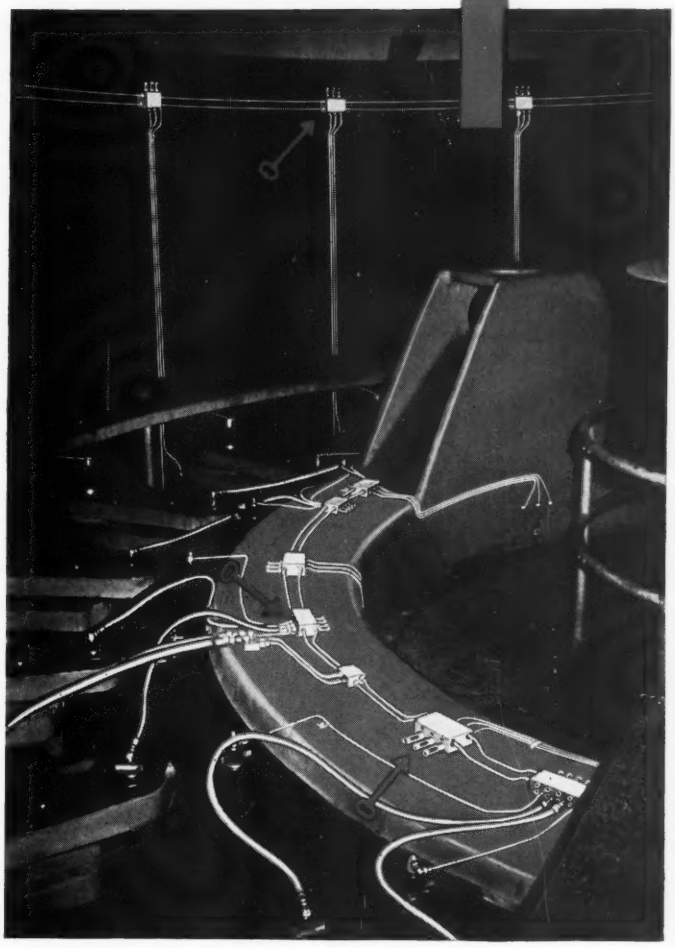

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